AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 41 No. 539. APRIL, 1951 PRICE : 2s. 6d.

AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Publishers: ILIFFE & SONS LTD., DORSET HOUSE, STAMFORD STREET, LONDON, S.E.1
Telegrams: Sliderule, Sedist, London **Telephone: Waterloo 3333 (60 lines)**

COVENTRY:	BIRMINGHAM, 2	MANCHESTER, 3:	GLASGOW, C.2
8-10, CORPORATION ST.	KING EDWARD HOUSE, NEW ST.	260, DEANSGATE	26B, RENFIELD ST.
Telegrams: Autocar, Coventry	*Telegrams:* Autopress, Birmingham	*Telegrams:* Iliffe, Manchester	*Telegrams:* Iliffe, Glasgow
Telephone: Coventry 5210	*Telephone:* Midland 7191-7	*Telephone:* Blackfriars 4412	*Telephone:* Central 4857

PUBLISHED MONTHLY—SECOND THURSDAY

Annual Subscription: Home and Overseas £1 17s. 6d., including the Special Number; U.S.A. $6.00; Canada $5.00

VOL. XLI. No. 539 APRIL 1951 PRICE 2s. 6d.

Torque Conversion

READERS who have followed our articles dealing with the design detail of the American fluid transmissions will doubtless also have studied the leading articles giving impressions of their performance in service. It seems to be generally agreed by those who have tried them, that in their present complex form, they are not completely suited to road and traffic conditions in this country. For the British automobile industry they nevertheless present a problem, and one that has broadly two distinct facets. Firstly, as a major export, must not the British car sooner or later provide at least a similar type of motoring if it is to retain its customers abroad? Secondly, what type of transmission layout will be forthcoming for this purpose? and this constitutes the major part of the problem. Whatever the scheme however, the first essential is that it must be very much simpler mechanically.

Basic Problems

With our relatively small outputs the cost of any scheme approaching the complexity of the American designs would be quite prohibitive. Simplicity is also essential because of the problem of servicing in the remoter areas to which British cars are now exported. The technical problem would appear to have again two facets, though basically it is still the original one that has always faced the petrol vehicle designer, namely that of securing varying torque at varying speeds from the fundamentally constant torque prime mover. Superimposed on this problem is that of furnishing this characteristic under a control either fully automatic or nearly so and demanding little more from the driver than the operation of a single pedal.

Up to the moment, attention has been directed entirely to the transmission system. In other words, designers have accepted the limitations of the orthodox power unit and its characteristics, and have concentrated their efforts upon devising such mechanisms as will best utilise or convert this power supply to the requirements of the vehicle performance. With some temerity we make the suggestion that the time may now have arrived when it may be wise to see if there are not still unexplored potentialities within the power unit. Now that engine size carries no tax penalty, the first obvious step in the solution of the general problem would seem to be to utilise the potentialities of considerably increased cubic capacities. Taken to an extreme, were sufficient power available, no torque conversion gear of any kind would be necessary.

Within more practical spheres a large engine with ample output, in combination with a "part time" fluid coupling and a two speed and reverse epicyclic gear would go a large part of the way. It would however involve some degree of fuel wastage, which is one of the chief criticisms now levelled at the highly evolved American converter transmissions. In all the American designs, both engine capacity and fuel are freely, if not lavishly, employed to attain the desired results. It would seem therefore that a profitable avenue for exploration lies in the study of ways and means of using fairly large engines more economically. Assuming an extremely simple silent low cost transmission as outlined above, a surge of power would be needed only to attain the higher cruising speeds in minimum time, and some fuel wastage would probably be unavoidable. Once under way however, a locked direct drive could be employed, and the power could advantageously be very much reduced.

The running costs of the large engine are of course now solely a matter of fuel, and means could surely be devised for linking fuel input much more closely to power requirements. A fuel injection system of suitable type giving the requisite mixture strength range might offer a solution. Stratification systems may embody potentialities. Alternatively, by entirely new approaches, the normal fuel metering process by carburetter might be made to function effectively as a power metering device.

A New Approach

The steam engine with its highly developed cut-off, expansive, valve gear, furnishes a simple technical parallel as to objective. Up to the present however, to confess even to speculations as to the possibility of designing some such characteristic into the petrol engine is tantamount to an admission of complete ignorance of first principles. The substantially constant torque nature of the petrol engine is admittedly a basic factor. It is nevertheless conceivable that by superimposing some other principle equally basic, the one may be induced to modify the other. In short, the engine should be regarded purely as a fuel burning device to be fed with precisely the minimum amount of fuel needed for the work of the moment, this work being done without appreciable loss of either thermal or mechanical efficiency. The world of science and engineering teems with the records of achieve-

ments involving far greater difficulty and infinitely greater originality.

Concurrently with this research, efforts should be made to eliminate all other sources of fuel waste. Despite the efficiency of the modern carburetter, there is still widespread and general fuel wastage. Over say, a year's running, much fuel is unnecessarily lost. About half-a-pint of petrol is thrown away every time a two-litre engine is started from cold. New methods of rapidly attaining normal carburation are still needed, despite the greatly improved warming up times of to-day.

It is unlikely that there will be any general return to small engines, so that much greater quantities of heat are now rapidly available. This should be applied to the mixture in the form of heat in air, immediately the engine is firing. As the bulk heat supply becomes progressively available, it should be employed in conjunction with carefully metered fuel throttling, linked directly to the power requirements of the moment at various running speeds. An over-riding control would be arranged so that full power became at once available when conditions demanded.

In short, the aim should be so to reduce the fuel penalty of the large engine at cruising speeds, that consumptions become no greater than the actual horse power requirements necessitate. Put in another way, what is wanted is a big capacity engine with ample power in reserve that will nevertheless give, say 40 horse-power only, a large part of the time as economically as a 1,000 c.c. unit. We have to discover how to make the petrol engine operate in such a way that, subject to mechanical losses only, the maximum proportion of the heat units supplied as fuel, are converted to work done, and over a wide speed range. In this way, part of the required torque variation would be economically provided by the power unit itself.

Rearmament

At the moment the full effect of the arms programme on the British automobile industry is a matter of speculation, but that it must eventually have adverse results is unquestionable. Already there are shortages of essential materials and the machine tool situation is rapidly becoming more difficult. Even now delivery of standard machine tools is delayed and it is probable that special purpose tools for purely civil purposes will soon be virtually unobtainable. This must mean that many of the ambitious programmes for improving production facilities will be slowed down, if not stopped completely. Responsibility for minimising the troubles that must be faced will fall heavily on two departments, production engineering and purchasing.

Despite the known difficulties the automobile industry is expected to continue its magnificent contribution to the export trade of the country. Every possible means, therefore, must be taken to avoid, or at least minimise, any further advances in the prices of vehicles, although every branch of the industry faces rising prices for all goods and services. Fortunately, although the machine tool situation may rule out any spectacular advances in production techniques through re-tooling, the major organisations have all carried through extensive re-tooling projects and the general level of equipment is high.

Much more serious difficulties are to be anticipated owing to the shortages of essential materials. For example, the shortage of sheet steel has already affected the industry. Other shortages must follow. Any alleviation of their effects through the employment of alternative materials must be a relatively long term policy. The immediate ill-effects may in some degree be minimised if a bolder policy is followed regarding contingency stocks.

Production executives generally, and understandably, follow a cautious policy in this matter with a marked tendency to call for too large rather than too small contingency stocks. A bolder policy in this respect would certainly help to ease matters during the initial stages of the introduction of the new programme, since the limitation of supplies of essential material will make a more rapid turn-over of stock a matter of major importance.

It is also to be hoped that purchasing departments will exercise greater restraint than they did in the period immediately following the 1939-45 war. Then there was what can only be called an orgy of over-ordering. Any suggestion of impending shortages naturally leads purchasing departments to go in for stock piling. On this occasion it is to be hoped that restraint will be shown. Otherwise there will be an increase in the number of governmental controls, with all the frustration and inconvenience they inevitably cause.

In normal circumstances exception would not be taken if a purchasing agent looked after the interest of his own company without any thought of what the reactions would be on other companies. But the times are not normal, and inevitably, much of the resources of the automobile industry will be diverted to the defence programme. It may be difficult, if not impossible fully to employ the resources that are left free for normal production. This is a case in which the interests of any individual company must be subordinated to the interests of a great industry. Great skill and resource will be called for if serious pitfalls are to be avoided. Even so the scale of the armament expenditure may well prove the final factor in wreaking irreparable harm to this country.

CONTENTS OF THIS ISSUE

P.N.D.L.R.

A Detailed Study of the American Torque-Converter Transmissions

By O. D. North, M.I.Mech.E.

(Continued from page 90).

Converter Design

General Motors, on Buick and Chevrolet, employ the "polyphase" system that they have developed. In this there are two stators, with vanes of different angles, mounted on roller and wedge free-wheels, the inner members of which are splined to a sleeve carried by the gearbox casing. The pump is also in two sections, the main or primary pump being preceded by a narrow set of blades called the secondary pump.

These are set at a greater helical angle than the entry of the primary pump blades and at low speeds the oil strikes the back of these blades. They are therefore driven faster than the primary pump, rotating on a roller free-wheel on the pump hub.

As the engine speed rises the secondary pump blades may be said to "catch up" with the oil and are driven by the free-wheel, thus forming a modifying extension of the primary pump blades. Both stators are in action at the beginning of conversion. but with rising speed the secondary stator is no longer impinged on by the oil stream, and therefore free-wheels. Finally, the primary stator free-wheels also, this being the end of the torque conversion, and the beginning of the fluid coupling period. This subdivision of the members gives an effect similar to that of adjustable blade angles and improves the overall efficiency of the converter considerably, especially at the change-point from conversion to coupling.

Packard employ one stator only, but two sets of turbine blades, both in use all the time, the second set being between the stator and the pump blades. The Packard converter has aluminium-alloy blades cast in plaster moulds to secure a smooth surface and maximum accuracy. A rounded form is given to the entering edges, which to some extent accommodates the incompatibility of oil flow with blade angle which it is the object of the General Motors "polyphase" system to reduce.

Ford use a simple converter, having pump, turbine and stator only. The pump and turbine blades are of thin sheet steel, but the stator is an aluminium alloy casting with thick, bulbous blades. Studebaker use a simple converter with pressed steel blades throughout.

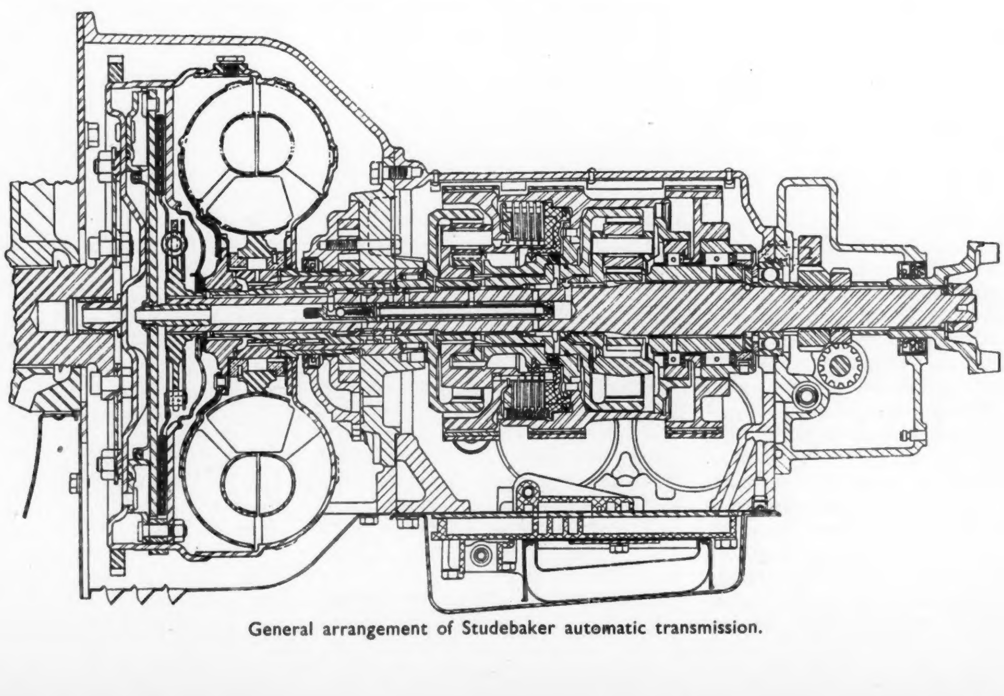
Converter Construction

Special interest attaches to the pressed steel construction of the blading. This is employed in one form by Chevrolet and, in a quite different form, by Ford and Studebaker. Chevrolet press the blades, in progressive dies, with narrow flanges conforming to the surfaces of the inner and outer members. The metal thickness is about 0.030″ and the blades are initially located by multiple spot-welding at the flanges, subsequent fixing being by copper brazing.

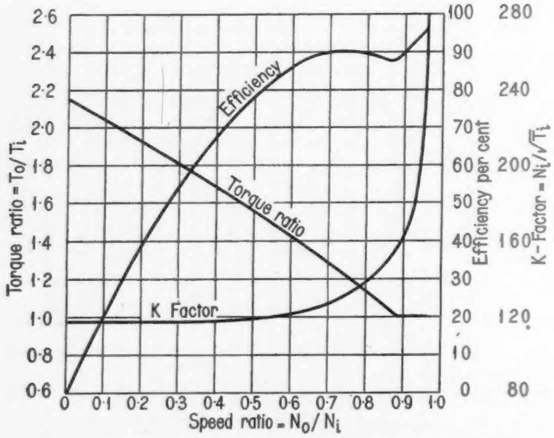
Studebaker and Ford, in their Borg-Warner converters, use fairly hard steel blades with tabs. Some are sprung into recesses in the supporting members and secured by a locking ring. Others have tabs passed through slots pierced in the pressed steel members and locked by rolling over the ends. Ford use an aluminium alloy casting for the pump body which forms most of the converter enclosure. It has radial ribs cast on its outer surface and these act as a fan and cooling surface for heat dissipation. The recesses for the pump blade tabs are die-cast in this case, but in the Studebaker converter they are half-pierced in the steel shell.

Stator Free-wheels

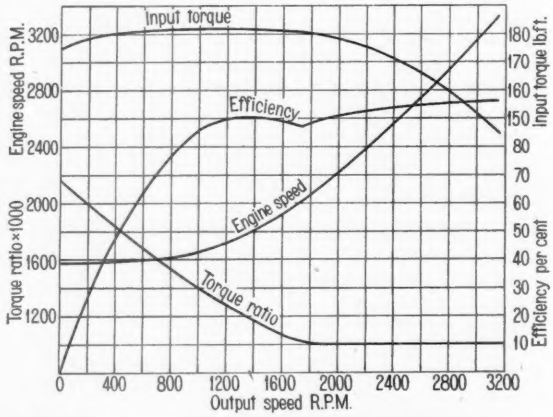
Buick and Chevrolet use roller free-wheels with the wedge-bottomed recesses for the rollers machined in the outer races. The intermediate lands form journal surfaces in combination with the inner races. Studebaker and Ford use what in the U.S.A. is



General arrangement of Studebaker automatic transmission.



Efficiency, torque ratio and K factor curves for Studebaker torque converter.



Full throttle performance curves of Studebaker three-element torque converter.

known as the "sprag" type free-wheel. In this, both inner and outer races are plain cylinders and the locking is effected by radial strips with slightly offset cam ends. These are biassed by coiled spring circlets fitting in recesses, or by double pierced cages, one inside the other, forced in opposite directions by springs.

The fact that the outer race is plain facilitates manufacture in the case of the Studebaker converter, which has a pressed steel stator assembly. The race is pressed into this and secured by pressing tabs into keyways machined in its outer diameter to take the torque. Since the foregoing operations are apt to upset the concentricity of the complete assembly, the bore and end faces of the free-wheel outer race are ground after final assembly, a procedure that would be impossible with a roller free-wheel.

Packard use a sprag free-wheel, but locate it in an unusual position behind the front oil pressure pump. The stator is splined to a running sleeve passing through the converter pump hub, the sleeve driving the front oil pressure pump. Unlike the roller type, the sprag free-wheels cannot serve as journals also. Flanged bushes are therefore pressed into the outer races on each side to carry the stators.

Converter Oil Supply

In all cases the maintenance of a pressure of about 30 lb. per square inch in the converter to prevent cavitation is the first duty of the oil pressure pump arrangements. A circulation of oil is afforded, fairly rapid in those cars using an external oil cooler and less rapid in Ford and Studebaker in which the cooling is done in the ribbed converter body itself by air flow. The supply of oil is through a port uncovered by a spring-loaded piston valve, generally forming part of the pressure regulation system. Final escape is by a spring-loaded check valve set to about 15 lb. per square inch, after which some, or all, of the oil goes to lubricate every bearing in the gearbox. The important point is that as the engine comes to rest, the piston valve closes and the check valve comes on its seat. This locks up the oil in the converter and prevents air entering while the car is standing.

In all cases the hub of the converter pump is extended as a sleeve into the gearbox casing. Its rearward end is tongued to drive the front oil pressure pump. Since there is at least one sleeve and one shaft within this (two sleeves in the case of Studebaker) the diameter of the outermost member is considerable. The duty thrown on the oil seal, which has to hold 30 to 40 lb. per square inch on a diameter of about 2" running at perhaps 3,000 r.p.m. with oil possibly at 300 deg. F., may justifiably be considered severe.

Gearboxes

The epicyclic trains of Buick, Chevrolet and Packard are almost identical in basic design. The input shaft from the converter passes through the epicyclic box and is splined to the driving sun gear. This, which has 33 teeth, meshes with three long planet pinions with 21 teeth, extending forward to mesh with three short pinions (30 teeth). These in turn engage a 27-tooth reaction sun wheel, which, when held stationary, gives a low gear of 1.82 to 1, the output being to the planet cage. The 30-toothed short pinions also mesh with an annulus having 87 teeth, which, when held, gives a reverse gear with a reduction of 1.64 to 1.

The figures given happen to be for the Packard, but both General Motors cars have the same trains except that the short pinions and the annulus have a few more teeth, giving a 1.82 reverse, while still keeping the 1.82 low ratio. This seems to be an excellent design, the very awkward ratio of 1.82 to 1 being obtained without elaborate compounding. Further, any desired reverse ratio can be had without altering the forward gear. Ford-Mercury go one better than this, getting three forward speeds (i.e., two indirect ratios) from the same number of gear-wheels differently arranged, by the addition of an extra friction clutch.

In the Ford arrangement the output is to the annulus and the planet cage can be braked if desired to give both low gear and reverse. Three long pinions mesh with the annulus, with a front sun wheel and with three short pinions which clear the teeth of the annulus but mesh with a small sun wheel in the same plane. There are brake bands to hold the front sun wheel and the planet carrier and clutches capable of driving either the small sun wheel or the large one, or both at once, which gives direct drive. For all forward speeds the clutch on the small sun wheel is held in engagement. Low gear is obtained by holding the planet carrier and intermediate gear by holding the front sun wheel and direct by holding both at once.

Reverse is obtained by disengaging the small sun wheel clutch, referred to as the "front clutch", and engaging the clutch connected to the front sun wheel, while at the same time holding the planet carrier.

The ratios are :—

Low	.. ..	2.44 to 1
Intermediate	..	1.48 to 1
High	.. ..	1.00 to 1
Reverse ..	..	2.00 to 1

In the Ford gearbox the two clutches are in front of a partition in the box which carries a bush supporting the front end of the planet cage. The two brake bands, one on the rear clutch drum and one on the planet cage, are thus well supported. It will be found

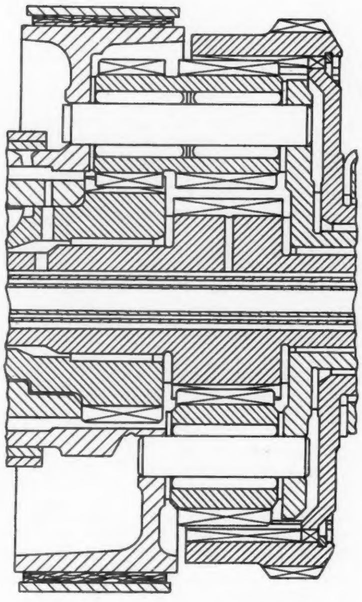
in all the gearboxes that this point has received special attention. This is because all the brake bands are of the single-anchor type and thus impose considerable side loads on the drums, especially the reverse drum. The designs, in this respect, are in marked contrast to the Wilson gearboxes. In these, all brake bands have double anchors taking the reaction as pure torque. A complicated assembly of a series of epicyclics thus functions with only the support given by very narrow bushes and wide-spaced shaft bearings.

The Studebaker gearbox, which also gives three speeds forward and a reverse, is on quite different lines. Here the torque converter is coupled to the annulus of a front planetary set. The planet cage is connected to the annulus of a second (rear) planetary, the cage of which is splined to the output shaft. The sun wheel of the front set can be clutched to its cage, locking it solid. It is also connected to a drum surrounding the rear set and closing in behind it on to a free-wheel clutch acting on the sun wheel sleeve of the rear set.

This sleeve also carries, on a second free-wheel, a brake called the "forward" band by the application of which the sleeve is prevented from reverse rotation. A brake, called the "reverse" band, can hold the planet cage of the front set while another brake acting on the sun wheel of the front planetary set is called the "low" band. The forward band is applied for all the forward gears. Top speed is direct by the engagement of a friction clutch in the converter. This connects the engine direct to the output shaft. The multiple disc clutch in the gear set is also applied, locking the front planetary, which revolves at engine speed driven by the converter turbine. Since the output shaft is also running at engine speed the rear planetary set is also in effect locked. The sun wheel sleeve of the rear set therefore free-wheels within the "forward" brake drum.

For intermediate gear no change is made except to release the direct-drive clutch. The converter turbine then picks up the load, after the engine has accelerated, through the rear planetary set, the sun wheel of which is held against reverse rotation by the free-wheel in the "forward" drum.

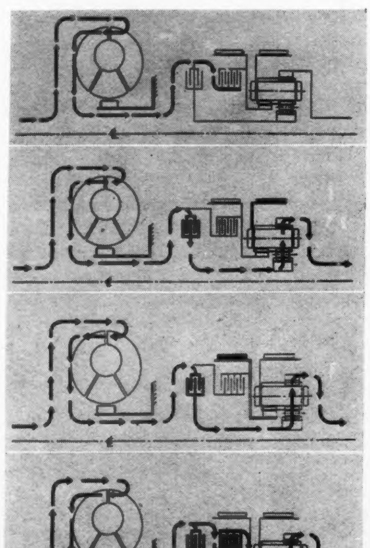
The free-wheel action eliminates any need for synchronisation of the release of the direct drive clutch with the engagement of any brake band, the relevant one, i.e., the "forward" band, being engaged all the time. A second, and not so desirable, consequence is that the engine is not available as a brake through the intermediate gear.



Ford-Mercury gear set assembly.

When the throttle is shut, however, the direct drive clutch is automatically engaged at all speeds above 12 m.p.h., so that the engine becomes available as a brake, but at unity ratio only.

A third feature of the double free-wheel arrangement is that when in intermediate gear the car will not run back on a hill since both the rear set sun and the rear set annulus (via the disc clutch) are held against reverse rotation by the two free-wheels. For low gear the disc clutch is released, unlocking the front planetary set, which takes up the drive to the rear set, its sun wheel being held by the application of the low band. Since the low band holds the outer race of the first free-wheel the sun wheel of the rear set is now held against motion in both directions and low gear is available for braking purposes, i.e., there is no longer any free-wheeling action.



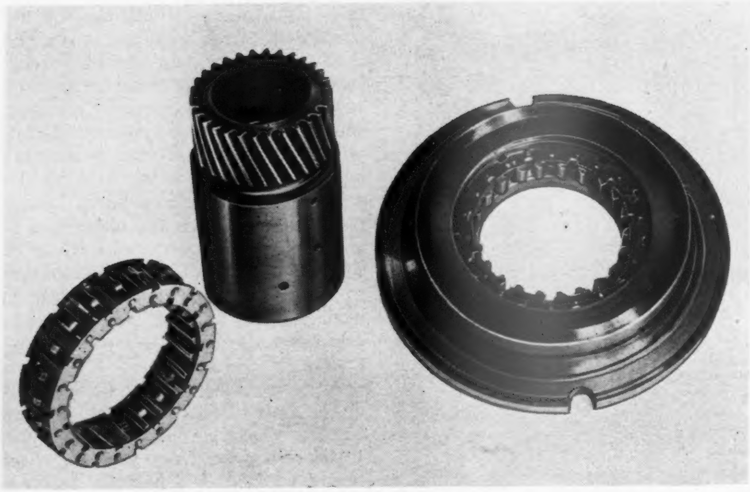
Ford-Mercury power-flow diagram for (top to bottom) neutral, low gear, intermediate gear, high gear and reverse.

For reverse gear all brakes and clutches are released except the reverse band. Since this holds the front set planet cage, the front set sun wheel runs backwards and picks up the rear set sun wheel and turns it backwards. Now the rear set annulus is permanently connected to the front set planet cage and is therefore held at rest. The rear planetary set, with a stationary annulus and a reverse-rotating sun wheel, drives its planet cage (splined to the output shaft) backwards, giving reverse movement to the vehicle.

The Studebaker is unusual in having anti-friction bearings on the axis of the gearbox, all the others using plain bearings. Packard and Chevrolet employ a ball bearing at the rear, beyond the main gearbox. In the Studebaker gearbox Torrington needle roller bearings carry the turbine sleeve with its integral annulus, and within this the throughgoing output shaft is carried in another pair of Torrington bearings. The rear end of the output shaft is carried on a ball bearing. The three brake drums are each supported, directly or indirectly, either on a long bush or on two well spaced short bearings. Any appreciable eccentricity or tilting of the epicyclic trains under brake-band anchorage loads is prevented.

Gear Train Details

Helical teeth are used throughout, the helix angle is generally 20 deg. and some pinions are induction-hardened. They all run direct on needle roller bearings. The planet pins are also induction-hardened. Some gears are crown-shaved, the long pinions being treated as two separate sets of teeth. On the Chevrolet, the reverse annulus and brake drum with web and hub are integral in high-grade cast iron. The two annuli in the Studebaker gearbox are steel forgings integral with their fairly long sleeves. The single annulus in the Ford-Mercury gearbox, which transmits the final drive to the output shaft, is a steel ring, with the parking lock teeth cut on its outer diameter. Its internal



Rockford sprag-type free wheels in Studebaker transmission.

teeth at their shouldered rear ends fit on to teeth cut in a flange on the output shaft, a circlip keeping the ring in place. End thrusts set up by the helical teeth are in general taken by loose bronze thrust washers. Where a ball bearing is fitted to the output shaft, as in Chevrolet, Packard and Studebaker, it gives end location to that part, and eliminates one thrust washer.

The planet cage, especially when combined with the output shaft, is an awkward part for quantity production. The most advanced design is to be found on the Chevrolet, in which a flange upset on the output shaft is spigoted and riveted to a pressing, like a small brake drum, having three holes punched to clear the planet pinions. To a flange on the front end of this pressing is spigoted and riveted a stout front plate stamping, bored to take the ends of the planet pins and having locally-hardened teeth cut in its outside diameter for the parking lock pawl.

Clutches

The direct-drive clutches incorporated in the converters on Packard and Studebaker only, are single-discs faced with moulded cork materials bonded to the disc. The smaller multiple-disc clutches in the gearboxes have either alternate steel and sintered bronze plates or alternate slightly dished cold-rolled steel plates and plates carrying bonded, moulded facings with a metal content. Spiral oil release grooves are cut in the facings.

Application is by direct-acting pistons sealed either by piston rings, as on Packard, or by synthetic rubber seals, as on the Chevrolet. They are supplied with oil by banjo connections sealed by piston rings with hooked ends to facilitate assembly. In the case of the Packard, sealing is effected by the close fit of the two bushes supporting the clutch drum on the rearwardly-projecting nose on the front cover of the gearbox. The two clutches used in the Ford-Mercury get their oil supply from banjo rings on the rear of the output shaft.

The front clutch on the Ford gearbox has to take the same torque as the rear one. It is however, always engaged when in forward gear and never has to pick up under load. It happens to save space to use a smaller piston and fewer clutch discs in the front clutch. The requisite grip is attained by means of a multiplying lever in the form of a Belleville washer, serving also as piston return spring.

The clutch operating pistons have a diameter of between 4 and 5 inches and it has been found desirable to fit bleed valves. These are opened by the retraction of the piston. They prevent the building up by centrifugal force of pressure within the rotating cylinder sufficient to cause partial engagement of the clutch when it is intended to be free.

Epicyclic Brake Bands

These are between $1\frac{1}{4}$ and $1\frac{1}{2}$ inches wide, split at one point, with recessed lugs into which fit struts. One of these goes to an adjustable anchor screw while the other is thrust by the gear servo cylinder either direct or through a multiplying lever. A very thin moulded lining is bonded to the band and is machined after fitting. The Buick brake bands exemplify the most advanced production method. Here a strip of cold-rolled steel is cut to length, has small welding projections struck on its ends in a press and is then bent into a circle. A short bent strip is then projection-welded to the outside across the junction and is afterwards copper brazed to make an absolutely intimate joint. The band is then disc-ground to width and stretched to size on an expanding mandrel.

The inside surface, which has no machining operation, is shot-peened. This gives a good base for the bonding cement and also, by setting up compressive stresses, gives the band a tendency to spring open slightly when the lug portion is sawn through as a final operation. After the band has been sized by stretching, the lining is bonded by electrical heat; it is then machined inside. The recesses in the lug member are machined and finally the band is sawn through. Normal brake material is sometimes used for the low gear bands, which do not have to take very high torques, but for the more heavily loaded reverse bands a lining incorporating metallic particles is generally employed. Wear of the bands is stated to be very slight and adjustment, after initial assembly, is

Studebaker rear planetary set, free wheels and rear brake drum.

not generally contemplated.

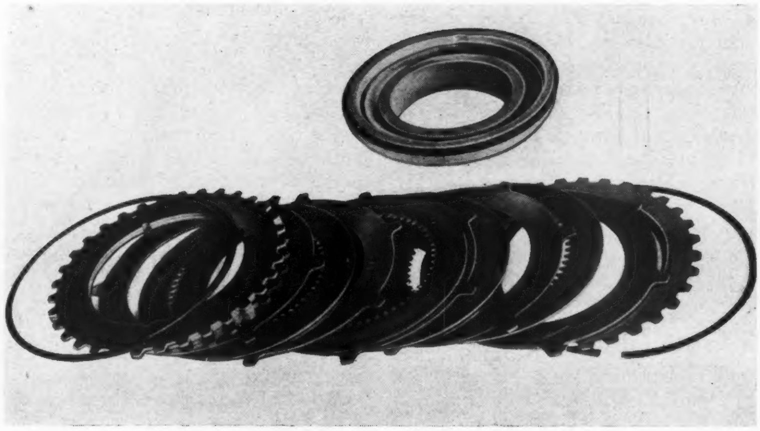
It would appear that one of the reasons for the use of the simple single-anchor band, apart from its low cost, is that it can be relied on to pick up more gently on the overrun, owing to the reversal of the "wrap-up" effect. One of the features of all torque-converter transmissions, as offered to the public, is that they can be changed down to low ratio by an incompetent driver at quite high speeds without severe shock. Many other features, some very complicated, are incorporated to assist in ensuring "foolproofness".

Oil Pressure Pumps

The hydraulic control system requires a considerable amount of oil and the pumps have also to cope with the steady drain of the converter feed, which, where an external cooler is fitted, is a considerable volume. On the Studebaker and Ford-Mercury with direct air cooling of the converter body the converter feed is much less, being only that required to deal with the lubrication of the gearbox bearings.

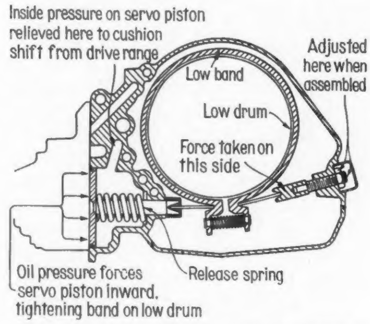
The front oil pump, driven by a sleeve forming part of the hub of the converter pump, has to be of large capacity for another reason. It is expected to supply ample oil while the engine is running at, say, 1,000 to 1,500 r.p.m. to operate forward and reverse bands in frequent succession. This need arises when the car is being manoeuvred into an awkward position or when it is being "rocked" out of a hole in snow or mud. The situation is aggravated by the fact that the reverse band generally requires a doubled pressure of about 180 lb. per square inch, so that leakage and pump slip are important factors. On this account the front oil pump has to be much larger than is necessary for normal steady driving, and, if special precautions were not taken, would waste a serious amount of engine power at high speeds.

All front oil pumps are of the internal gear type with crescent abutment, except in the case of Packard,



Components of Studebaker multi-plate clutch.

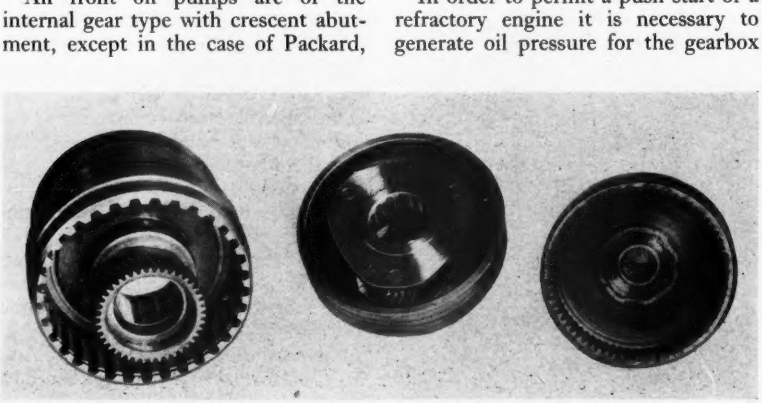
who use the Gerotor pump, in which no abutment is necessary. The seal is made by the tips of the rotor teeth against the tips of the teeth on the internal gear. The pump pinions generally have 25 teeth, 9 D.P. with a big pressure angle to avoid interfer-



Chevrolet "Low" band servo mechanism.

ence with the internal gear. The face width is about $\frac{5}{8}$ in. Normal working pressures in "Drive" range are from 80 to 90 lb. per square inch. These are in some cases reduced with partially closed throttle, as not being required to transmit the reduced torque, so that the pump load is reduced.

In order to permit a push start of a refractory engine it is necessary to generate oil pressure for the gearbox



Studebaker multi-plate clutch housing and front planetary set.

when the front engine-driven pump is at rest. A similar pump, but of about half the capacity, is therefore mounted on the output shaft, except in the case of Studebaker, which has a spur-geared rear pump driven by spiral gears. Advantage is taken of this second smaller pump to bypass the front pump as soon as the car has attained sufficient speed for the rear pump to take over.

This occurs at speeds of from 25 to 45 m.p.h., according to the temperature and viscosity of the oil. General Motors, in Buick and Chevrolet, arrange the cutting out of the front pump by means of a spring-loaded pressure regulating valve, fed by both pumps through check valves and having a land bypassing the front pump separately. As speed increases the land on the valve opens the front pump bypass until a point is reached at which it is generating no pressure. The two pump check valves are, in the Chevrolet, ingeniously combined in one unit, consisting of a hairpin spring made of flat strip, each of its straight ends seating on one of the pump delivery holes.

Packard, who use a friction clutch locking out the torque-converter on direct drive, cut off the feed to the converter when the clutch is engaged and the cessation of converter hydraulic losses makes cooling superfluous. This, by reducing the oil demand, makes it possible for the front pump to be cut out at a lower speed than would otherwise be the case.

That the power taken by the oil pumps if both are at work at high speed can be serious was shown by comparative tests made by Studebaker, both with and without front pump bypassing, a reduction in transmission temperature of 30 deg. F. being shown. Dynamometer tests showed the two pumps together to be taking over 2 h.p. at times. In some of the

systems requiring 180 lb. per square inch pressure to hold the "Low" range gear band, possibly about three horsepower may be wasted on a steep hill, where the low car speed will not allow the rear pump to deliver enough oil to overcome the slip and leakage losses. The large front pump is therefore working at high speed and high pressure.

The oil used in a torque-converter is of very low viscosity, in order to reduce the hydraulic losses. Gear pumps of the form used, having a very low "aspect ratio" and considerable circumference open to end leakage, are hardly suited to develop relatively high pressures at low speeds. Possibly this is the reason for the use of a normal spur gear pump at the rear of the Studebaker transmission.

The above comment, however, has little application to the large and powerful vehicles in question. Their performance is so good that "Low" range is practically never required. The point is, however, that considerable caution is needed in any application of the American designs to small, light vehicles in which fuel economy is a vital point and the power-weight ratio poor.

Hydraulic Fluid

The following extracts from a paper read before the S.A.E. in November, 1949, by H. R. Wolf of General Motors Research, and J. L. McCloud of the Ford Motor Co., on "Automatic Transmission Fluid (type A) for Passenger Cars", are of interest.

After emphasising that automatic transmissions cannot satisfactorily be operated with standard engine oils and the like, the paper gives the qualities necessary for the purpose.

Apart from the more obvious properties, such as low viscosity, high viscosity index, low pour point and high flash point, stress is laid on resistance to oxidation. This must be considerably higher than in normal engine oils. The effect of the oil on synthetic seals at high temperatures is specially mentioned, together with anti-foaming properties. Chatter or "squawking", which are "stick-slip" phenomena, require special emphasis on the frictional characteristics of the fluid and its liability, after prolonged use, to form films upsetting the smooth operation of the oil-wetted clutches and brakes.

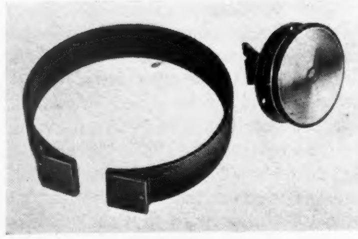
A part of the approved specification is as follows :—

Viscoscity, Saybolt Universal 210°F. . .	54 sec. min. 56 sec. max.
Viscosity index . .	150 min.
Flash point	365 deg. F.
Fire point	395 deg. F.
Pour point	—35 deg. F. max.

Hydraulic Control

Buick and Chevrolet differ from the rest in that no automatic engagement of any clutch or brake has to be provided. On Packard and Studebaker there are automatic converter "lock-up" clutches and on Ford, automatic operation of both gear clutches and gear bands. Buick, as described in the "Automobile Engineer" for May, 1950, use a spring-loaded pressure regulating valve, controlling first the supply to the converter and moving on to act as a selective relief valve, venting the front pump before the rear one.

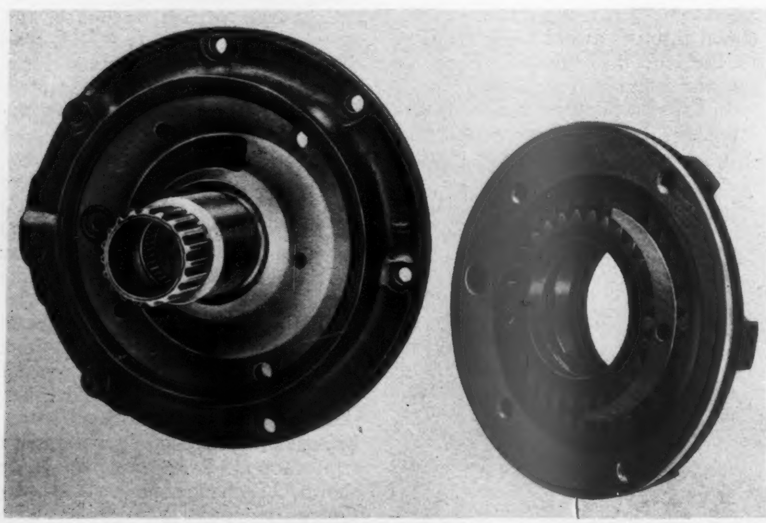
Normal pressure in "Drive" range is set at about 90 lb. per square inch, and the rear pump takes over at about 45 m.p.h. The manually-operated shift control valve admits line pressure to areas on the regulating valve. These areas act in support of the calibrated springs and double the regulated pressure when in "Low" range and Reverse, giving some 180 lb. per square inch. From the particulars given in the maker's excellent service manual it would appear that the rear pump is incapable of keeping up this high pressure and that both pumps are therefore in operation all the time the car is in "Low" range. The power absorption may be as much as 3 h.p. at 40 m.p.h.



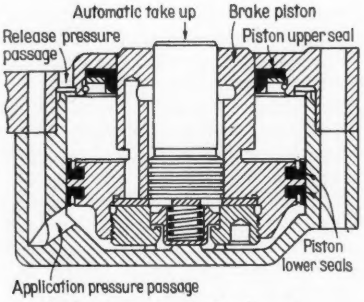
Studebaker brake band and operating piston.

Buick continue to use the "accumulators", or spring-loaded cushions, in the supply lines to both direct drive clutch and low gear servo. In low gear the device is without the ball check valve shown in the "*Automobile Engineer*" article. Of particular interest is the omission, from current models, of the complicated anchor piston (shown on page 171) with its control ports. Equally sweet change-over from "Drive" to "Low" range has now been obtained by calibrated orifices in the low gear accumulator. Another change concerns the piston of the reverse servo, which now operates the gear band lever through a stiff compression spring, giving a cushioning effect.

Chevrolet use a similar regulating valve but with variable pressure loading applied to it in quite a different way. In "Drive" range the line pressure acts on two areas, one always connected and the other via the direct drive clutch feed. The line pressure acting on these areas balances with two compression springs to give a pressure of 43 lb. per square inch. A vacuum diaphragm connected to the induction pipe is loaded with a spring which adds to the load on the regulating valve to give a further 47 lb. per square inch, making a maximum of 90 lb. per square inch in "Drive" range. Eighteen inches of manifold vacuum entirely removes this additional load. Hence the actual line pressure is considerably reduced in normal running at part throttle and reduced torque. Oil leakage and flow to the converter being thus cut down the average duty of the pumps is



Studebaker front (engine driven) oil pump.



Packard reverse hydraulic unit.

much reduced. The large front pump starts idling at relatively low speeds while the rear pump is working at reduced pressure.

A pressure of 180 lb. per square inch is still required in "Low" range, the increase coming in two parts. First, in low the clutch is vented and one of the hydraulic forces tending to open the regulator valve is removed. This gives 78 lb. per square inch pressure. Starting from this point the pressure is still further built up by admission of line oil, via an "accumulator" cushion, to a floating piston. This acts as a strut between the vacuum diaphragm and the lever loading the regulating valve. It brings the regulated pressure up to 180 lb. per square inch, pushing back the diaphragm spring so that it no longer plays any part. There is thus no vacuum modulation of line pressure in "Low", because full pressure is required to hold low band against engine braking on a hill, since the band is then "unwrapping". The Chevrolet system omits the high gear "accumulator" used on Buick, but uses the cushioned pressure rise for both low and reverse gears. The reverse piston is spring-cushioned also.

The tricky change-over from "Drive" to "Low" is ingeniously assisted by connecting the clutch cylinder feed to the inner face of the low band servo piston. On the application of control pressure to the outer face, the low band grips the drum, but with a much reduced force until the pressure in the clutch feed has dropped. Simultaneous full engagement of high and low gears is thus made impossible, and without any intermediate pause with both disengaged which could cause engine racing.

Buick, in common with all makers except Chevrolet, arrange the control and regulating piston valves in two aluminium alloy die-castings They are superimposed in a horizontal plane with a metal diaphragm between them Suitable holes in the diaphragm line up with passages cast in the upper and lower elements so that complicated cross-connections can be made without any pipework. Aluminium alloy is used, both for ease of diecasting and machining and because of the absence of gritty particles in the cast surfaces liable to jam the delicate regulating valves, etc. It has, however, been found that particles arriving from outside are apt to cause trouble by becoming embedded in the relatively soft metal.

Chevrolet manage to combine manual control and pressure regulating valves, together with the single "accumulator", in an iron casting secured to the rear face of the bell housing. It incorporates the nose-piece that supports and feeds the rotating "Drive" range clutch cylinder. Since this member is only separated by a diaphragm from the front oil pump, considerable simplification of oil passages can be effected..

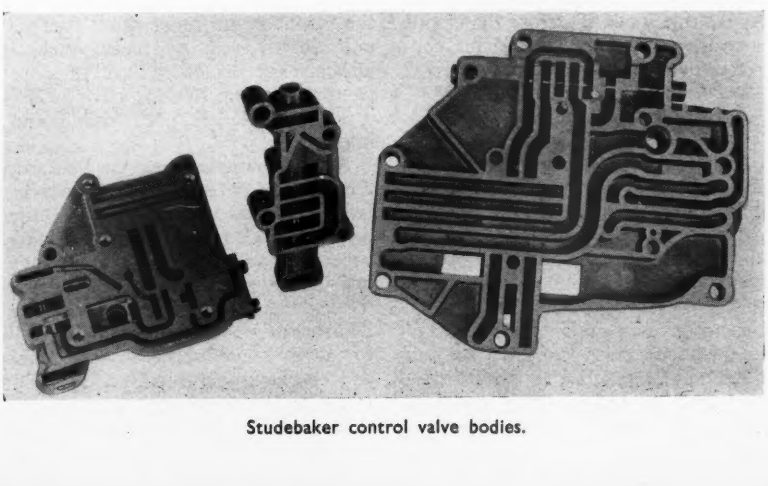
The castings, together with others in the transmission, are heat-treated in a continuous furnace 40 feet long, with a maximum temperature of 1,200 deg. F. to relieve all stresses. They are then transferred to a salt bath at 900 deg. F. which dissolves all traces of core sand in about two minutes. Subsequent washing, acid dip, rinsing etc., leave the castings absolutely clean and free from any siliceous material. Tool life is greatly increased, machining is facilitated and the reliability of the finished product improved.

Another interesting point in Chevrolet machine-shop practice is the wide use of lapping machines for finishing the flat faces of the valve bodies where oil-tightness is essential. Flatness is the objective, a high degree of surface finish not being necessary since gaskets are employed. Multiple lapping machines take 12 pieces at a time. They have many special features, including continuous conditioning of the cast-iron lap. A slow, precision operation is thus converted into one dealing with an output of 100 units an hour.

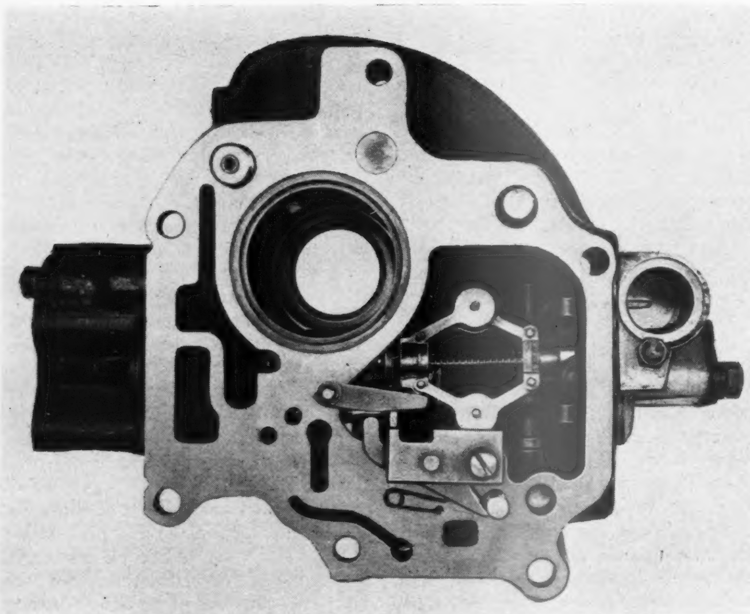
In the Packard transmission the low and reverse servo pistons are of large diameter and work through multiplying levers as well. The large displacement of oil which this would normally involve in taking up band clearance is avoided by the incorporation of small automatic take-up plungers in the main servo pistons. These, fed by check valves from the lower side of the pistons, act like automatic tappet adjusters. They eliminate all band clearance with a gentle pressure before the main piston starts to move, it actually being held for an instant by oil on the upper side which is vented as application begins.

Thus, the main piston only moves enough to take up elastic spring of the parts. Enough hydraulic and mechanical advantage can therefore be provided to hold the "Low" range band without increasing the line pressure above the normal 85 lb. per square inch. The duty of the oil pumps in low range is very greatly reduced, not only by reduction of working pressure but also because the reduced pressure makes it possible for the small rear pump to deal with the requirements at fairly high speeds.

Boost is, however, necessary to hold the reverse band and is obtained roughly in the same way as in the case of Buick. Line pressure is admitted to an area on the "pump selector valve", which is really the regulating valve, normally at atmospheric pressure. Reverse pressure is 180 lb. per square inch and the servo is the same size as that for "Low" range. Reverse having a ratio of 1.62 to 1 and low 1.82 to 1, the torque on the gear drums will be as 2.62 is to .82, or as 3.2 to 1. There is, however, no need to provide so much grip on reverse as the figures would indicate, since this gear is always engaged with the car practically at rest and at small throttle openings. The oil film is therefore broken down before the full engine torque comes on. The actual valving on the Packard is so exceedingly



Studebaker control valve bodies.



Studebaker gearbox rear end casing showing governor.

complicated that it is not even fully clear in the 11 large pages of letterpress and 13 illustrations employed in the service manual. The mechanism includes a centrifugal governor driven from the output shaft, i.e., proportional to car speed. This modulates a small tapping from the main line to give pressures varying from 29 lb. per square inch at 15 m.p.h. up to 61 lb. per square inch at 56 m.p.h. Another valve, termed the "throttle valve", is really a reducing valve loaded by a spring compressed by accelerator pedal movement. This gives about 25 lb. per square inch on the tick over up to 60 lb. per square inch at full throttle.

The "governor" and "throttle" pressures are opposed in operating a valve, the "direct drive clutch shift valve" which controls the engagement of a cork-faced single disc clutch inside the torque converter. This locks the converter solid down to about 15 m.p.h. with throttle closed. Above this speed therefore the engine is available as a brake without converter overrun slip on both "Drive" and "Low" range.

When the accelerator is depressed at low speeds the "throttle pressure" beats the "governor pressure" and the clutch is disengaged. At higher speeds it requires more depression of the pedal to free the clutch and beyond about 50 m.p.h. no amount of acceleration will unlock the converter. At this speed it would make no contribution to engine torque and would only be wasting power as a slightly slipping fluid coupling. At part throttle the car stays in locked drive down to quite a low speed, but the clutch is instantly released on giving full throttle thus preventing "pinking". A tapping of the direct drive clutch line goes to another piston valve, the "converter feed valve", which cuts off the circulation of oil through the converter when the direct drive clutch is engaged.

For the gearbox, the usually manually actuated valve controls the drive range clutch and the low and reverse gear bands. Supply to the first two is under the further control of a "timing valve", in which a spring-loaded piston moves, shuttle-fashion, at a speed regulated by a calibrated orifice. This controls the timing of the change up or down. A further valve, called the "modulating valve", is operated by "throttle pressure" to modify the pressure supplied to engage the "Drive" range clutch according to engine torque.

There are two more piston valves; one is a double-ended shuttle-acting "pump check valve" and the other is the "front pump relief valve". This is loaded to 15 lb. per square inch at all times, supplemented by oil pressure from the "pump selector valve" when it is desired to bring the front pump into action.

Basically simpler than the Packard, the Studebaker control has no "modulation" of line pressure by throttle position. It also has no "timing valve", because a free-wheel covers the change over from direct drive to intermediate. This condition is in some respects equivalent to the change from "Drive" to "Low" range on the Packard, since a step-change is involved and the action takes place at speed.

Cushioning of the application of all brakes and clutches is provided by a "pressure dome" or air vessel in the main pressure line. The opening of any valve supplying a clutch or brake piston causes a momentary drop in pressure in the whole system. The air in the vessel expands somewhat, serving afterwards to cushion the return to normal pressure as the piston completes its travel. Line pressure is about 80 lb. per square inch for all operations except the engagement of reverse band. For this 200 lb. per square inch is given by applying line pressure to an area on the front pump relief valve to supplement the spring pressure.

Converter supply is by a spring-loaded piston valve holding back the 80 lb. per square inch line pressure to give 27 lb. per square inch. Since the converter has its integral cooling arrangements, the oil flow is only that leaking through the various clearances to provide lubrication. With only this limited escape the building up of 200 lb. per square inch line pressure for reverse would give 147 lb. per square inch in the converter, probably bursting the casing and destroying the seal. This danger is evaded by taking a connection from the reverse servo line to a piston that applies additional load to the converter feed valve. This holds back 173 lb. per square inch instead of 53 and maintains the converter pressure at the normal 27 lb. per square inch.

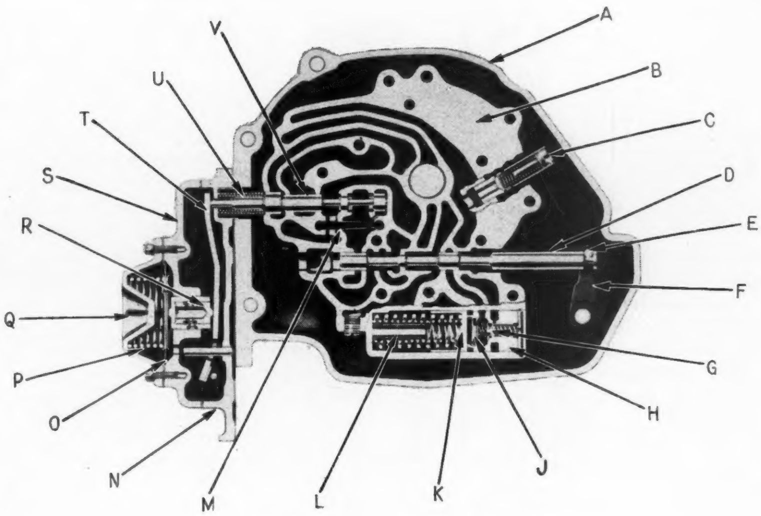
Operation of the piston valve controlling the direct-drive clutch is by direct mechanical connection to a fly-ball governor with a compression spring between the upper and lower collars. The upper one works the valve while the lower one is moved up and down by a straightforward mechanical connection with the accelerator pedal. The clutch valve is opened by the falling of the top collar of the fly-ball governor. This occurs with increased car speed, as the balls fly out, or by closing of the engine throttle, which lowers the whole assembly, or by any combination of the two movements. A rather subtle detail is the provision of a tiny plunger working parallel with the piston valve and loaded by the clutch operating pressure when this comes on. This "feed-back" makes the control slightly unstable. In other words, the clutch has a definite tendency to stay in or out, instead of dithering at the change-over.

The servos are all direct-acting, the "forward" band having a single piston and the low and reverse bands two pistons in tandem. The reverse pistons are very large and would draw a considerable amount of oil from the line in taking up the clearance. This is avoided by initially supplying oil to

one piston only and letting the other one suck its oil from the sump, until the rise in pressure as the band begins to tighten opens a spring-loaded shuttle valve. This admits line pressure to the second piston and simultaneously closes its suction passage. The shuttle valve also plays a part in getting a quick release. As soon as the pressure has dropped it opens the second cylinder to the sump again, discharging its oil direct and not via the control valve.

Two ingenious little devices are included, namely a parking pawl interlock piston and a reverse interlock piston valve. This vents the reverse servo line to atmosphere until the rear pump pressure has dropped to zero by the forward motion of the car ceasing. The first gives the advantage that the parking lock teeth need not be rounded to prevent accidental engagement when the car is moving. When parked on a gradient the locking pawl will therefore not be forced so hard against the "dead-centre" mechanism which holds it in position and will therefore be easier to withdraw. Cases have apparently occurred of the locking pawl becoming jammed beyond the possibility of normal manual withdrawal under such circumstances. The reverse interlock seems to be a highly desirable feature for obvious reasons.

Working pressure in the Ford-Mercury system varies from 75 lb. per square inch under cruising conditions up to 125 lb. per square inch on low gear and 150 lb. per square inch in reverse. No sudden boost is applied but pressure is regulated by hydraulic means. Two modulating pressures are applied, one by a variable reducing valve connected to the engine throttle control and the other by a centrifugal governor on the output shaft. This governor is of extreme simplicity, consisting of a piston-weight mass moving radially in a die-cast cylinder bolted to a stamping, with an opposing



Chevrolet control valve body.

A. Transmission case.
B. Valve body.
C. Pressure relief valve.
D. Manual valve.
E. Manual valve pin.
F. Manual valve lever.
G. Accumulator valve.
H. Accumulator valve body.
J. Accumulator check valve.
K. Accumulator piston and springs.
L. Piston stop.
M. Dual check valve.
N. Servo cover.
O. Vacuum diaphragm.
P. Vacuum diaphragm spring.
Q. Modulator cover.
R. Hydraulic modulator pistons.
S. Modulator housing.
T. Modulator control lever.
U. Pressure regulator valve springs.
V. Pressure regulator valve.

balance weight, pressed on to the output shaft. Oil connections to the governor and to the two gearbox disc clutches are by a three-groove banjo connection, with piston-ring seals just behind the governor.

In "Drive" range the Ford system changes automatically from high gear, or direct drive in the gearbox, to intermediate. This change is made by a valve subject to the opposed "throttle" and "governor" pressures as in the Packard system. The "governor" pressure also acts on another valve preventing the transmission shifting into low gear at speeds above 40 m.p.h. Manual selection of "Low" range transfers the automatic control to intermediate and low gears in the gearbox. This control however is not subject to "throttle" pressure but to "governor" pressure only. Low gear is held until the car speed reaches 40 m.p.h. and the change down occurs below this speed regardless of accelerator position. This is an important feature, since low gear is vital for engine braking. Were throttle control applied, as in "Drive" range, the gearbox would change automatically into intermediate when the accelerator was released. The servo for the front band of the epicyclic gearbox has to hold only about engine torque at converter stall and the loads are small enough to permit cylinder and multiplying lever being made as light-alloy die castings. The rear band has to hold reverse torque, about six times engine torque, and requires a larger cylinder and a stamped steel multiplying lever giving more purchase.

'British Standards' Exhibition

THIS year the British Standards movement attains its Golden Jubilee, and as a part of the celebration an Exhibition supported by practically the whole range of British Industry will be held at the Science Museum, South Kensington, during the two weeks beginning 18th June, 1951.

The Exhibition will also show how research at one end of the production chain, and quality control at the other, are linked with and helped by standardization. Other special features will include apparatus used in testing for compliance with British Standards. The President of the Board of Trade will open the Exhibition at 11.30 a.m. on the 18th June. Admission will be free, and opening hours will be 10 a.m. to 7 p.m. each day (except Sunday) from 18th to 28th June inclusive. (1949)

Bearing Lubrication

ALL engineers concerned with the design, production, or maintenance of machinery will be interested in *Bearing Lubrication* a publication issued by C. C. Wakefield and Co. Ltd., Grosvenor Street, London, W.1, at a price of 21s. 0d. The book covers a much wider field than is indicated by the title. The reason for this is that although proper lubrication starts on the drawing board, good design can be spoiled by faulty machining, bad fitting, inferior materials, or by delivering to the bearings wrong lubricants in incorrect quantities. The first three chapters deal briefly with the theory of friction and lubrication, properties of lubricants, causes of deterioration, and calculation of bearing loads, including loads imposed by various forms of gearing, such as hypoid gears, etc. Later chapters deal with plain, ball and roller bearings, grease lubrication and oil circulating systems, as well as the special problems concerning vertical spindles, slides, etc. In addition there are sections describing oil and grease sealing devices, and bearing and lubricating system troubles. The book is very well written, with emphasis on the practical aspects of lubrication, and is illustrated with extremely clear diagrams. (1944)

CAST MILLING CUTTERS

A B.S.A. Development for Greater Efficiency and Lower Costs

IT would appear that the limits of efficiency have now been reached for cutting tools of the conventional tungsten and molybdenum high speed steels, although there are certain surface treatments that give increased efficiency. Overall machining efficiency can however be obtained by reducing the cost of the tools. Such reductions can be effected in several ways. One that is of considerable interest is the use of milling cutters produced by a casting process. B.S.A. Tools Ltd., Mackadown Lane, Marston Green, Birmingham, have found that such tools lead to considerable savings.

The problem of producing cast milling cutters has been approached by B.S.A. Tools Ltd. primarily from the point of view of tool-makers rather than foundrymen. Consequently, cutter design and material have been the governing factors, and full advantage has been taken of the fact that cutters of excellent design for cutting, for example, staggered tooth side and face and face types, can be produced as easily as cutters of less good design such as straight tooth side and face cutters. In order words the maximum advantage can be taken of the latest developments in cutter design without the increase in production costs that would have to be met in the manufacture of cutters by conventional machining methods.

In addition, the material used is based on a 1.3 per cent. carbon, 13 per cent. chromium alloy containing small amounts of tungsten and vanadium. The advantages of using this type of alloy steel during the present shortages of tungsten and molybdenum are obvious. This steel has excellent casting properties with complete freedom from massive segregation in the cutting teeth, provided due precautions are taken and the correct casting technique is used. Good and bad microstructures from the teeth of similar side and face cutters are shown in Figs. 1 and 2 respectively. Both are ×100 magnifications. It will be noted that the spheroidised structure shown in Fig. 1 is similar to that shown by a normal hot-worked 18/4/1 high speed steel and no traces of any cast structures are present.

These cutters are capable of removing stock at a rate at least as great as that of a standard cutter made from 18/4/1 high speed steel. It is however, absolutely essential that cast cutters be run with a copious supply of "soluble" type cutting fluid. Tables I and II give typical operating conditions for cast side and face and face mills respectively. Where the machine tool in a given category, light, medium or heavy, is very rigid and backlash eliminators are incorporated, the feed rate may be increased one step if the component set-up is sufficiently rigid.

In the production of these cutters the nature of the pattern, the composi-

Table I
Typical operating conditions for side and face cutters (4in. diam., ½in. to ¾in. wide)

Steel up to 38 Tons/sq. in.				Steel from 38 to 54 Tons/sq. in.			
Cutting Speed: 95 R.P.M. (100 F.P.M.)				Cutting Speed: 86 R.P.M. (90 F.P.M.)			
Depth (ins.)	Feed in Ins. per Min. for Light Machines	Medium Machines	Heavy Machines	Depth (ins.)	Feed in Ins. per Min. for Light Machines	Medium Machines	Heavy Machines
0.20	7½	9	—	0.20	4½	6	—
0.40	6	7½	—	0.40	3¾	4½	—
0.80	4½	6	—	0.80	3	3¾	—

Steel from 54 to 70 Tons/sq. in.				Cast Iron up to 180 B.H.N.			
Cutting Speed: 76 R.P.M. (80 F.P.M.)				Cutting Speed: 67 R.P.M. (70 F.P.M.)			
Depth (ins.)	Feed in Ins. per Min. for Light Machines	Medium Machines	Heavy Machines	Depth (ins.)	Feed in Ins. per Min. for Light Machines	Medium Machines	Heavy Machines
0.20	3	3¾	—	0.20	9	12	—
0.40	2⅛	3	—	0.40	7½	9	—
0.80	1¾	2⅛	—	0.80	6	7½	—

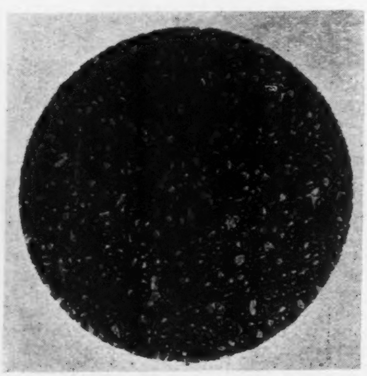


Fig. 1. A good micro-structure of material used for B.S.A cast tools.

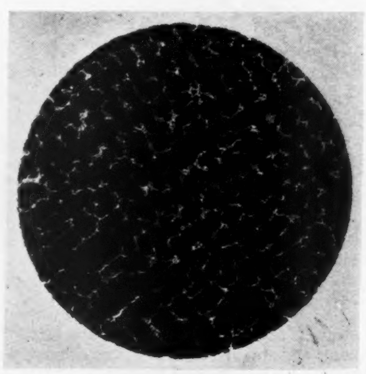


Fig. 2. A bad micro-structure of material used for B.S.A. cast tools.

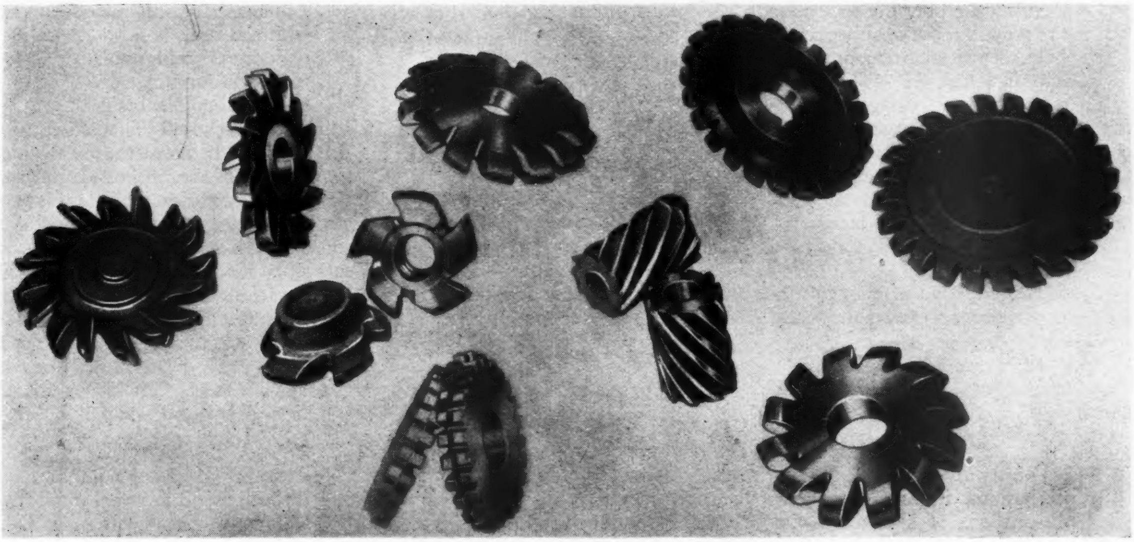


Fig. 3. Typical B.S.A. cast cutters.

tion of the mould, the composition of the tool steel and the method of pouring are carefully controlled. They differ greatly from normal foundry procedure. The design of the pattern and even more particularly of the mould are of primary importance and call for care and attention to a number of details if sound castings are to be produced. However, there is virtually no tool shape that cannot be produced by this process. A representative selection of cast cutters is shown in Fig. 30.

Advantages of the Process

Among the advantages claimed for tools made by this process are :—

(1) The tool designer has full scope to design tools of maximum efficiency without the limitations imposed by conventional methods of machining from the solid.

(2) It is possible to produce milling cutters with alternate teeth having opposite helix angles with no more difficulty than a cutter with identical teeth.

(3) Cutting teeth can be produced having any shape in any direction. It is not necessary to generate the backing or support of a tooth as one, or a series, of straight lines.

(4) Cutters can be produced with ample swarf clearance and still be efficient after a number of re-grinds.

(5) Where a cutter has a constant axial shape and plain ends, as for example a helical slab mill with plain ends, it is possible to produce from one standard pattern cutters of any length within the limits of the pattern. A pattern for a 6in. long cutter will produce tools of any length from ½in. to 6in.

(6) Where form relieved cutters are made it is possible to take advantage of the special B.S.A. Tools Ltd. surface treatments such as Golden Arrow and Blue

Table II
Typical operating conditions for face mills (3in. dia.)

Steel up to 38 Tons/sq. in. — Cutting Speed: 120 R.P.M. (95 F.P.M.)				Steel from 38 to 54 Tons/sq. in. — Cutting Speed: 107 R.P.M. (85 F.P.M.)			
Depth (ins.)	Feed in Light Machines	Ins. per Min. for Medium Machines	Ins. per Min. for Heavy Machines	Depth (ins.)	Feed in Light Machines	Ins. per Min for Medium Machines	Ins. per Min for Heavy Machines
0.10	7½	9	12	0.10	4½	6	7½
0.20	6	7½	9	0.20	3¾	4½	6
0.30	4½	6	7½	0.30	3	3¾	3¾

Steel from 54 to 70 Tons/sq. in. — Cutting Speed: 95 R.P.M. (75 F.P.M.)				Cast Iron up to 180 B.H.N. — Cutting Speed: 86 R.P.M. (67½ F.P.M.)			
Depth (ins.)	Feed in Light Machines	Ins. per Min. for Medium Machines	Ins. per Min. for Heavy Machines	Depth (ins.)	Feed in Light Machines	Ins. per Min. for Medium Machines	Ins. per Min. for Heavy Machines
0.10	3	3¾	4½	0.10	9	12	14½
0.20	2⅛	3	3¾	0.20	7½	9	12
0.30	1¾	2⅛	2⅛	0.30	6	6	7½

Arrow. As only the rake faces are ground these treatments will help to prevent wear on clearance faces.

There are many advantages pertaining to staggered tooth cast cutters in comparison with machined straight tooth side and face cutters. They include the following :—

(1) Higher speeds and feeds and greater depths of cut may be employed.
(2) Shearing action of chip formation reduces wear on the teeth, thus prolonging the cutter life.
(3) Less power is required for equal cuts.
(4) Metal can be removed at an average rate 70 per cent. greater while retaining a life equivalent to that of standard straight tooth cutters.
(5) The lower cost of B.S.A. cast cutters.

The range of the process is not confined to milling cutters. Other items that have been produced include:

(1) Blanks for manufacture into thread rolls.
(2) Special turning tools.
(3) Inserts for broaches.
(4) Extrusion die blanks.

In addition to supply finished cutters, B.S.A. Tools Ltd. can supply castings to members of the Milling Cutter and Reaming Association. Since this process eliminates all milling operations, soft machining operations are necessary only for producing the hole and keyway, the cutter being finished by grinding after heat treatment.

SIMULATED ROAD TESTS

An Engine Dynamometer Control for Fuel Evaluation

ROAD tests involving full-throttle acceleration in high gear with observations of knocking tendency of the fuel throughout the acceleration are complicated by such uncontrollable variables as ambient temperature, wind direction, traffic hazards, etc. According to A. R. Isitt, M. R. Wall and A. G. Cattaneo, in an *S.A.E. Preprint*, November 9-10th, 1950, the electronic device ERL (electronic road load) improves the accuracy of fuel evaluation by applying to the laboratory-installed engine a dynamometer load substantially identical to that experienced by the road engine. The control reproduces the combined inertia and resistances of a car in high gear regardless of events before and after, by simple operation of the throttle as in a car on the road.

A full-throttle speed-time curve was first obtained for a 1946 Chevrolet standard coupé on the road. Duplication of this curve was then attempted in the laboratory by proper design of the velocity and inertia channels in the dynamometer control. The new device consists in essence of a closed servo-system in which a speed-sensitive tachometer generator on the test engine shaft provides a signal to the control circuit, and thence to the amplifier supplying field current for the dynamometer to load the test engine. The control also has a velocity, an acceleration and an "error" channel.

Knock tests were made in a 1942 Chevrolet engine with iso-octane and normal heptane as reference fuels, and five fuels made up from straight run, thermal-cracked and catalytically cracked components. Each test fuel was examined with and without the addition of 1.5 c.c. T.E.L. per gallon. Modified Uniontown road knock tests in which the car was repeatedly accelerated from 10 to 60 m.p.h. under full-throttle conditions were duplicated by laboratory tests with the ERL dynamometer control by accelerating the engine from 500 to 3,000 r.p.m. Reproducibility of results with the control proved comparable to that with the road tests. Borderline knock tests on the road and in the laboratory with the ERL control showed that reproducibility was better with the latter.

The ERL control is concluded to be a practical means of rating motor-vehicle fuels in full-scale engines in the laboratory. The hazards and variables of road testing are reduced, and time and labour saved. With adjustments, the control can be used to apply the appropriate load to any engine, and with the inclusion of a simple throttle programme timer, to duplicate any desired operating cycle. (*M.I.R.A. Abstract No.* 5236).

The Influence of Stress on Corrosion

IN *Corrosion*, August and September 1950, the effects of stress on the internal structure and energy characteristics of metals are discussed by J. J. Harwood with relationship to their influence on corrosion reactions. The nature and importance of residual stresses and the non-homogeneity of worked metals are emphasised. Recent concepts of the nature of grain boundaries are reviewed and their importance in reactions where stress and corrosion act in a conjoint manner is described.

A review of the literature reveals that stresses (either by applied loads or of a residual nature) may influence the nature, rate and distribution of corrosion reactions in several weys: (*a*) by increasing the internal energy level of the metal system and causing a possible shift of electrochemical potential in a more active direction, (*b*) by causing an intrinsic increase in the rate of corrosion, (*c*) by damaging protective surface films, (*d*) by influencing polarisation reactions, (*e*) by changing the metallurgical characteristics of the metal system in promoting phase transformations, precipitation, etc., (*f*) by accelerating the rate of corrosion by purely mechanical effects.

The exact influence of stress on rates of general corrosion is still questionable, but does not appear to be of major consequence. The most serious effects of stress are in localised corrosion phenomena such as stress-corrosion and corrosion fatigue. Stress-corrosion of alloys is particularly discussed and the influence of metal composition and structure, environment, state and degree of stress is presented. Practically all known alloy systems can be made to crack from stress-corrosion in appropriate environments.

The oxide film, mechanical and electrochemical theories of stress-corrosion cracking are reviewed and it is shown that the experimental evidence favours an electrochemical mechanism. However, the exact mechanism of cracking may vary from one metal system to another and no theory presented thus far is adequate to account for all observed phenomena. Stress-corrosion cracking of alpha brass, stainless steels and magnesium alloys is still not understood clearly.

Methods of protection against stress-corrosion cracking include the use of barrier coatings, stress-relieving heat treatments, proper design and fabrication procedures, cathodic protection and the introduction of surface compressive stresses as by shot peening, rolling, or swaging.

[M.I.R.A. Abstract No. 5123.]

NOZZLE RECONDITIONING

Service Equipment Developed by Leslie Hartridge Ltd.

THERE is little doubt that many road transport vehicles fitted with oil engines operate at much less than optimum efficiency because there is a failure to carry out periodic examination and if necessary, reconditioning of the injector nozzles. An injector nozzle as produced by the maker is a precision component that has to operate under very onerous conditions. It must atomize fuel that may be dirty, and maintain the delivery of precise quantities of fuel into the combustion chamber at every injection. Further, at the needle seat there will be a contact stress in the order of 13 tons per square inch at each injection, that is about 600 times per minute. Wear, and with it, loss of efficiency, are inevitable under such conditions, but with regular servicing and reconditioning it is possible to restore the injector to a near approach to its original condition.

Hitherto many of the smaller operators have neglected injection nozzles because of the expense entailed, and many of the larger operators have failed to deal with the subject on a logical basis for want of proper equipment. To-day, there is available a relatively inexpensive equipment that has been specially developed to allow nozzle reconditioning to be carried out in a logical and practical manner.

Some three or four years ago Leslie Hartridge Ltd., 9 Victoria St., London, S.W.1, decided to extend their range of equipment for testing nozzles and fuel pumps by developing a nozzle grinding and lapping machine suitable for bringing the seating on both the needle and body back to original accuracy. In the course of the development work on this machine it was found that merely to produce a machine that would grind and lap efficiently was not sufficient. Other factors also had to be considered and finally the new Hartridge method of nozzle reconditioning was evolved.

The logical sequence for reconditioning an injector nozzle body and needle is :—

(1) Dismantle. (4) Clean.
(2) Examine. (5) Test.
(3) Recondition.

It was therefore decided that in

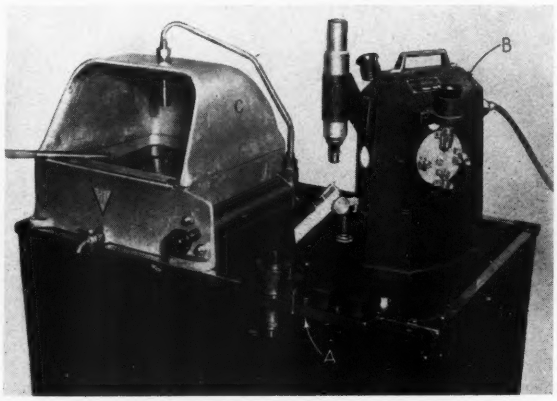


Fig. 1. Part of the equipment for the Hartridge method of nozzle reconditioning. A—dismantling and assembly jig, B—inspection microscope, C—cleaning cabinet.

addition to the grinding and lapping machine, the necessary equipment for the other operations should also be made available.

It is obvious that before reconditioning can be carried out the nozzle must be dismantled. In this connection it is only necessary to say that Leslie Hartridge Ltd. have developed a simple form of dismantling and assembly jig suitable for all types of

Fig. 2. The Hartridge grinding and lapping machine.

injector nozzles. It is shown at A in Fig. 1. The necessity for examination before reconditioning is attempted is not always realised, and much time and labour are spent on trying to recondition nozzles that are beyond recovery. To allow examination to be carried out expeditiously and effectively the universal microscope shown at B in Fig. 1 has been developed.

For examining needles this instrument is fitted with a microscope of either $\times 6$ or $\times 10$ magnification, according to preference. The needle is carried in adjustable vee blocks mounted on a slide that can be adjusted to any desired angle. Rough focusing is quickly effected by movement of the microscope in its sleeve and fine focusing by means of a micrometer screw beneath the fixture carrying the needle.

There are two examination stations for nozzle bodies, one for pintle type and the other for closed type bodies. All types of bodies in general use can be mounted for examination. A common light source is mounted in the body of the instrument. It is so arranged that a ray of light is projected to the bottom of the body seating to give perfect illumination. To facilitate examination, a watchmaker's eyeglass is fitted at each of the body stations.

The Hartridge grinding and lapping machine, illustrated in Figs. 2 and 3, has been developed to give results that hitherto have been considered impossible except from very expensive machine tools. Essentially it comprises a grinding head and a lapping head driven from a common motor. Complete freedom from vibration is essential in work of this nature, and as a first step towards ensuring this condition the motor is insulated from the remainder of the machine by three rubber mountings.

Drive from the motor to the grinding head is by rubber belt and to the lapping head by crossed flat belt. The grinding head design is based on the most advanced precision machine tool practice. Specially designed plain white metal bearing, similar to those used in the latest design of thread rolling machines,

carry the grinding head spindle. This special type of bearing in conjunction with the very good lubrication system gives the smoothest possible running. In addition it ensures the maintenance of original accuracy over very long periods of service.

The arrangements for holding and driving the needle or lap are particularly interesting. To begin with the barrel of the workpiece is mounted in two vee blocks. Between the vee blocks there is a small circular rubber driving belt which at the end remote from the work is carried over a pulley mounted on a shaft arranged to take one end of a flexible drive. Simple movement of a cam action lever pulls the circular belt back to hold the work securely in the vee blocks. The flexible drive shaft is attached to the lapping chuck and to the shaft carrying the driving pulley for the small circular belt which is under just sufficient tension to drive the needle or lap. This arrangement has advantages over the more conventional method of direct drive to the needle or lap through the flexible drive shaft. For example, it ensures that there is complete freedom from any tendency for the work to ride up or down in the vee blocks according to the direction of grinding wheel rotation. It also completely eliminates the possibility that the work axis will run eccentrically. Location of the needle or lap is taken from the bottom edge of the barrel, that is, from the point nearest to the element to be ground. Accurate angular setting for grinding the needle or lap is of fundamental importance. It is effected by means of a dial gauge that gives direct reading to one minute and

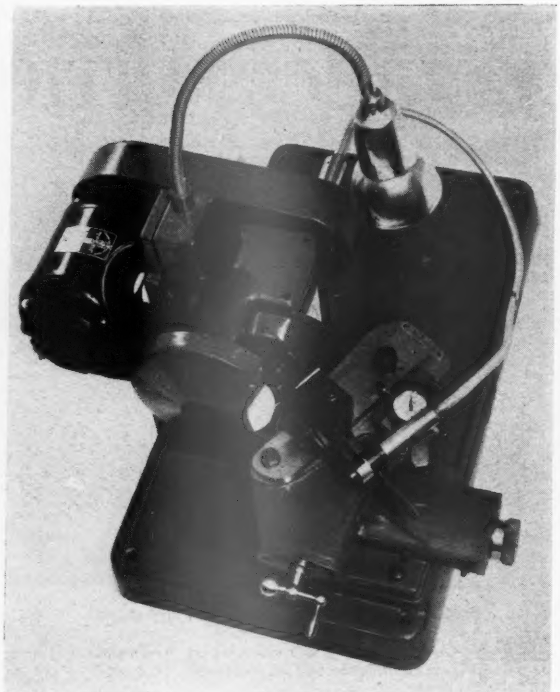


Fig. 3. The grinding head and angular slide.

allows estimation to within one-half minute.

A quick-action chuck is fitted to the lapping spindle. Laps can be inserted or withdrawn without there being necessity to stop the spindle. The lapping operation on a nozzle body is, in itself, simple, but great care must be taken to ensure that only the seating is lapped while the barrel diameter remains untouched. Should any lapping occur on the barrel there will be excessive back leakage. To facilitate lapping two grades of Hartridge lapping compound are put up in tubes so that it is a simple matter to apply the tube to the end of the lap in such a manner that there is no danger of any compound getting on to the barrel.

In connection with the lapping facilities, it is interesting to note that as a result of their wide experience in all matters relating to reconditioning injection equipment, Leslie Hartridge Ltd. consider that a needle can be reconditioned on this machine by grinding only and consequently needle lapping is practically eliminated. If needle lapping is considered to be necessary a range of female laps can be supplied.

Thorough cleaning of the needle and nozzle body after reconditioning is of great importance. The needle can easily be cleaned, but it is not so easy to ensure that the nozzle body is absolutely free from lapping compound. To facilitate cleaning, the equipment for the Hartridge reconditioning method includes a pressure cleaning cabinet, shown at C in Fig. 1. It is hand operated and is designed to direct a stream of cleaning liquid into the nozzle body and on to the seat under considerable pressure. Injector holding tools are supplied with the cabinet. Finally, every nozzle should be thoroughly tested after reconditioning. Tests may be carried out on one of the three Hartridge nozzle testing machines. These comprise a bench model, a cabinet model with an illuminated spray chamber and a motorised model. All three were fully described in *The Automobile Engineer* for March 1950.

This complete range of reconditioning equipment can be an important factor in the maintenance of diesel vehicle operating efficiency. Leslie Hartridge Ltd. gratefully acknowledge that in developing the equipment and the method they have had full co-operation from component manufacturers, particularly C.A.V. Ltd., engine manufacturers and operators.

NICKEL AND CHROMIUM PLATING

In view of the shortage of nickel which has developed recently, a special meeting of Committees of the British Standards Institution concerned with the preparation of Standards relating to electroplated coatings of nickel, decided that B.S.1224, Electroplated Coatings of Nickel and Chromium on Steel and Brass, the only Standard at present published in relation to such coatings, should not be amended. Instead a memorandum has been issued drawing attention to the provisions of the standard, by which it is arranged that the amount of nickel used will be reduced.

The memorandum is as follows :—"In view of the present shortage of supplies of nickel, and the importance of making the most efficient use of the supplies available, attention is drawn to the definition of the significant surface for plating on page 5 of this British Standard, which enables the manufacturer, by agreement with the purchaser, to reduce the surface to be coated in accordance with the standard.

"Attention is also drawn to the Note, on page 7, relating to standard classifications Ni 8S and Ni 5S, according to which the minimum deposit thickness may be a composite deposit of nickel and copper, provided that the final deposit of nickel is at least 50 per cent. of the whole." (1950)

COLD STARTING OF OIL ENGINES

A Simple C.A.V. Device for Low Temperature Working

THE starting of the compression ignition oil engine under low temperature conditions has been a matter of considerable importance to users from the early days. The problem involves not only the engine combustion system, the type of fuel and the method by which it is supplied to the combustion chamber, but also the related factors of the supply of the air necessary for combustion, the means by which the engine is cranked until it fires and continues to run under its own power, and engine lubrication. The cranking of the engine depends on the starter, the capacity of the battery, and the torque required to overcome the internal resistance of the engine. There are obvious limits to the size of battery which can reasonably be used, and therefore to the time for which the engine can be cranked. It will be obvious that however good the starting characteristics of the engines may be from a combustion point of view, this would be useless unless the power of the starter were sufficient to enable it to turn the engine at a speed high enough for firing and running. The power required can be reduced very considerably by the use of low viscosity lubricants. Battery performance at low temperatures has been substantially improved in recent years, and this, coupled with the use of the suitable lubricants referred to, has enabled the cold starting characteristics of the engine to be fully developed.

Before 1938, with the needs of the home market chiefly in view, it had been generally accepted that an engine which started promptly at freezing point, 32 deg. F. (0 deg. C.), would meet most Service requirements and was adequate for all general needs. When, in 1941, Russia entered the war, it became necessary to consider lower temperatures. Following the development of suitable cold-room facilities for testing all types of engine and vehicle, it was decided to provide for starting at —40 deg. F. (—40 deg. C.) for equipment intended for Russia, and 0 deg. F. for supplies for the continent.

To start a direct injection oil engine at 0 deg. F. without special aid necessitates cranking speeds up to as high as 280 r.p.m., and starting depends on combustion chamber design, fuel injection equipment, and the fuel used. Such speeds were impossible to attain under the starting conditions specified, and it was therefore necessary to con-

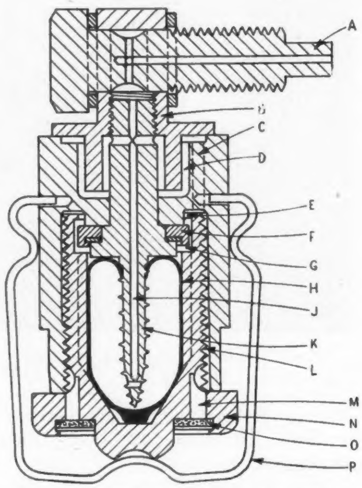


Fig. 1. Arrangement of C.A.V. cold starting device.

sider other aids which would ensure satisfactory starting. These included the provision of means for heating the inlet air and for the use of ignition promoting agents.

The use of an ignition-promoting agent offers by far the greatest scope for development of successful cold starting. For some time the use of various promoters has been known and practised by introduction into the induction air during the cranking operation. A great deal of work has been carried out on a very wide range of substances, including amyl nitrite, ethyl nitrate, and diethyl ether, for use as ignition promoters. The conclusion to be drawn is that diethyl ether is by far the most effective.

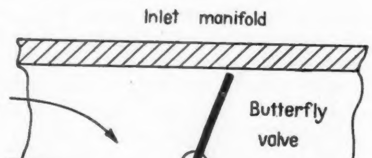


Fig. 2. Installation of starting device and butterfly valve in manifold.

The use of ether was successfully tried out in Service operations under severe conditions during the last war, and in view of the excellent results achieved, it was decided to adopt this method for commercial use in the C.A.V. cold starting device.

The function of the ignition-promoting agent is to provide a mixture which will ignite spontaneously at a lower temperature than the normal fuel-air mixture. The heat of reaction raises the temperature of the fuel-air mixture to such a value that it will burn.

As the ignition process is repeated, the amount of promoter required to sustain combustion of the fuel oil is reduced, until finally, as the engine has gained sufficient heat, normal firing proceeds without any aid from the promoter.

Two features should be fundamental in the design of the successful device, to effect reliable starts and to use a minimum quantity of ether, viz :

(a) Means of providing a rich ether-air mixture (termed "starting ether") during the first stages of cranking the engine, to obtain the first ignition.

(b) After first ignition has occurred, the rate of ether supply must be controlled to enable the engine to continue to run, so that the combustion chamber temperature is raised sufficiently and quickly for running on normal fuel oil only to take place. This "weak mixture" may be termed "running ether" or "running mixture".

A sectional view of the starting device is shown in Fig. 1. The actual height overall, including the bowl clip, is $3\frac{3}{4}$in. The device is inserted and held in the induction manifold by means of the screwed plug (A), the inner end of which projects into the manifold in way of the pitot passage through the butterfly valve (see Fig. 2). The body of the device (B, Fig. 1) is formed at the top as a lug, through which the screwed coupling plug passes. The starting fluid used, containing a high percentage of ether, is introduced into the bowl of the device by inserting a plastic capsule (H) containing 7 cc. of starting fluid and screwing up the bowl. This action causes the screwed central prong K to pierce the capsule, and enter it.

The bowl (N) is provided with a

coarse pitch thread to provide quick action with no danger of crossing threads. The spring clip (P) is provided merely to prevent the bowl becoming unscrewed and lost.

The central prong is drilled and cross drilled to provide a passage for the starting fluid into the upper portion of the body, in which is machined an annular chamber (D). The top of the passage is in communication with the outlet passage through the screwed plug and the lug, which forms a banjo type connection.

The centre prong bears a collar which carries a flexible washer (F), while the upper edge of the bowl has a rim (E). When the bowl is being screwed up, the capsule is pierced, but no upward flow of starting fluid takes place until the rim (E) comes into contact with the flexible washer (F), thus closing the bowl. Then, as the bowl is screwed further on to the body, the reduction in volume forces the liquid up the centre hole (J) and into the annular space (D). The dimensions are so arranged that this space is filled by the time the rim passes the sealing washer and re-establishes atmospheric pressure in the bowl. The tightening up of the bowl then seals the lower joint face; a felt air-filter (O) in the base of the bowl permits the entry of air by drilled holes (M) to the space (G) above the capsule, by way of a slot (L) cut through the thread. A further hole (C) connects the space (G) with the annular space (D), and thence with the outlet.

When ready to start, the bowl is unscrewed, the old capsule removed and a new capsule dropped into it.

Fig. 3. Inserting capsule containing starting fluid.

The bowl is then replaced, and screwed up home, the spring clip being finally swung into position.

With the butterfly valve in the idling position, the engine starter is then energised, a depression being caused at the pitot tube of the governor and at the outlet pipe of the cold starting device. The capsule having been pierced during the screwing up of the bowl, and the annular chambers above it filled with starting fluid, the depression created draws a very rich ether/air mixture from these spaces into the induction manifold, thus enabling the engine to fire immediately.

As the engine gathers speed, the depression at the outlet pipe increases. The first few strokes are sufficient to exhaust the relatively small volume of starting fluid from the annular spaces (D) above the bowl. After this has gone, the depression then draws ether from the capsule via the central passage (J), and this mixes with air entering by way of the air filter (O), holes (M), slot (L) and holes (C) to the space (D). The supply of fluid is such that the engine fires vigorously at the start while the rich supply of ignition-promoter fluid lasts, after which time the weaker mixture is sufficient to continue to assist ignition until normal, unassisted running on fuel oil only is established.

It should be noted that the "starting fluid" in the upper passages and annulus requires a slight depression only in the manifold to overcome its small static lift; the "running fluid", however, requires a somewhat greater depression owing to the increased static lift involved. Thus the first flow for starting is easily induced and is rich, while the secondary flow requires more air flow and is weaker.

The starting sequence is not under the operator's control in the sense that he can modify it or change the amount of fluid used. Where a pneumatic governor is fitted the engine is started without depressing the accelerator. As soon as the engine is running, the accelerator may then be operated normally. Where a mechanical governor is used, a butterfly has to be fitted, with separate hand control, and this is "closed" for starting (a suitable hole being provided in it to give the correct depression in the manifold), and opened when firing has begun.

DETERMINING DRAG OF AN AUTOMOBILE

A METHOD of determining the various drag components of a car is given, with an illustration of its application to a 1948 Plymouth coupé, by S. F. Horner, in *Automotive Industries*, Vol. 103, No. 6.

The car was accelerated to a suitable speed and then taken out of gear and allowed to coast freely along a smoothly paved highway. The speed was read from a calibrated speedometer and the deceleration from a device indicating deceleration by the height of a column of fluid. The total drag at different speeds was then found by multiplying the decelerations in terms of gravity by the effective weight of the car, which included the static weight plus the equivalent of the moment of inertia of the wheels. This extra weight was found by a pendulum method.

This total drag consists of an aerodynamic component, the rolling drag of the tyres, and the mechanical drag. The latter at no load was determined by deceleration tests of the wheels alone with the car on a jack and an allowance for load was estimated from handbooks as being independent of speed and equal to 0.001 times the weight of the car.

At very low speeds the aerodynamic drag is negligible and the measured drag is therefore tyre plus mechanical drag, and once the latter has been determined, the tyre drag at low speeds can be found. A graph of variation of this with tyre pressure is given. The tyre drag at other speeds was then calculated from formulae obtained from drum tests, the approximate formula for conventional tyres is $K_T=K_{TO}\,(1+\frac{2.5}{10^4}V^2)$ where K_T and K_{TO} are the drag load ratios at zero and V m.p.h. respectively.

After measuring and calculating all the mechanical and tyre drag components, the aerodynamic drag was found by subtraction to be approximately 0.032 times the square of the velocity in m.p.h. and the corresponding non-dimensional aerodynamic drag coefficient was then 0.52 for the car with the windows and cowl ventilation closed. Wind tunnel tests give coefficients of 0.2 for streamlined vehicles with faired undersides and 0.7 for passenger cars of the 1920 design.

The deceleration method is convenient for determining drag differentials due to modifications. An example is quoted in which the effect on the coefficient of opening the windows and cowl ventilator was found to be 0.02, which agreed with theoretical computations. (*M.I.R.A, Abstract No.* 5212).

FOUNDRY MECHANIZATION*

A Survey of Developments During Recent Years

IN this paper, the innovations that have taken place in foundry practice since 1942 are discussed, with special reference to the mass production of British Railways at Horwich, light, medium and heavy foundries. The advances that have been made in the semi-mechanization of jobbing foundries are also discussed. It may be stressed that there is need for mechanization in the jobbing foundry because of the scarcity of the skilled labour on which it is dependent.

Mass Production

Modern mass production practice is illustrated by the plant layout for railway chairs or railway baseplates and for brake blocks at the British Railway foundries at Horwich. For railway chairs or railway baseplates there are two mould conveyors with twelve pairs of machines. Each pair of machines comprises one roll-over machine for dealing with the drag or lower part of the mould, and a straight draw machine for the flat cope or top part of the mould. Both types of machine are hydraulically operated and fitted with under-sand frames.

Two castings of chairs are moulded in each box, and the estimated output is seven railway chairs per mould conveyor per minute. The weight of molten metal necessary to feed each box is approximately 100 lb. Therefore to give the output of castings, with two castings per box, 700 lb. of metal per minute is required. This equals 42,000 lb. or nearly 20 tons of metal per hour for railway chairs alone.

Five pairs of machines are installed to feed the brake block production side of the plant. Each pair of machines comprises one roll-over hydraulic moulding machine, fitted with under-sand frames, for making the bottom part of the mould, and one hydraulic straight-draw, similarly fitted up, for the top part of the mould. Probably three small, or two large, castings will be made in each box, and the estimated output from the single mould conveyor and from the ten moulding machines is an average of seven brake blocks per minute.

The weight of molten metal is calculated at 100 lb. per mould. Therefore with two castings per box approximately 350 lb. of metal per minute will be required for the envisaged production. This is about half that required for the chairs. For both plants 30 tons of metal per hour will be required to give a production of fourteen railway chairs and seven brake blocks per minute. Eight cupolas are installed. Four are rated to give up to 20 tons of molten metal each per hour, and four to give up to eight tons.

Knocking-out Station

A novel knocking-out station is

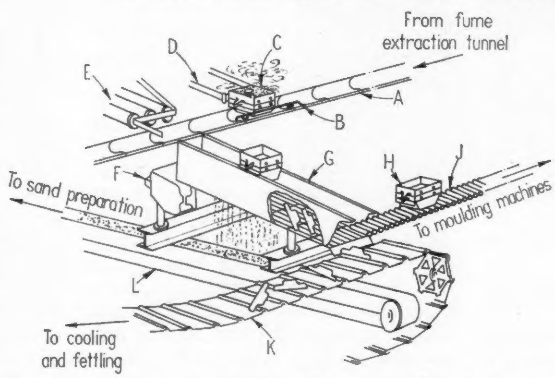


Fig. 1. Knock-out on British Railways mechanized foundry.
A—Mould Conveyor. B—Locating Strip. C—Moulding Box. D—Pre-pusher. E—Main Pusher. F—Vibrator. G—Vibrating Trough. H—Empty Box. J—Power Track. K—Hot Casting Conveyor. L—Belt Conveyor.

shown diagrammatically in Fig. 1. It originated at the Railway Executive's London Midland Region and has been worked out in conjunction with the Railway Company's engineers. This mechanized knocking-out station eliminates many of the inconvenient operations associated with knocking-out mass-produced castings. It comprises a mould conveyor, a pre-pusher and a main pusher, as well as a vibrating trough, the empty box return and the hot castings conveyor. It is usable only where the castings are of a very similar nature.

Moulding Boxes. The moulding boxes are all made from high-tensile steel. They are of welded construction and are all of the same dimensions, namely, the top part 20 × 18 inches by 4 inches deep and the bottom part 20 × 18 inches by 8 inches deep. The bottom part has tapered sides for lateral ramming during moulding with under-sand frame, it also has special skid bars welded to the sides for locating the boxes in the vibrating trough, on which are skid guides, and also for protecting from wear the bottom face of the box. The boxes are extremely accurately made, with ground joint surfaces, welded lugs fitted with hardened steel bushes, and clamping arrangements, with loose accurate clamps fitted on to two spigot accurately located into the sides of the top and bottom box parts. The bottom part has no bars. This allows the sand and castings to fall through the bottom when the box is subjected to vibration on the open vibrating trough.

Mould Conveyor. This type of mould conveyor is fitted with a complete table top composed of cast-iron plates, fitting into each other. The average speed of the mould conveyor is calculated at 10ft. 6in. per min., but the drive is fitted with a speed variator, giving 3/1 variation, or, from 6ft. per min. up to 18ft. per min. For the chair casting section each mould conveyor will handle 210 boxes per hour, and will give an approximate cooling time of 22 minutes.

In the brake block section, the speed of the conveyor is the same, and, consequently, the cooling time is the same, but here there is a variation due to the fact that only five pairs of moulding machines will be in operation against six pairs on each of the chair conveyers; and again there will be two, three, or even four, brake blocks per box, according to the size of the castings, whereas, on the chair plant, two castings are always moulded in each box. Hence, for practical purposes, the output of both mould conveyors on the chair castings can be calculated at 210 boxes per hour, and with two mould conveyers, 420 moulds per hour, which, with two castings per box, means a production of 840 chair castings per hour from both mould conveyers, or fourteen chair castings per minute.

The Pre-pusher. The pre-pusher is installed for the sole purpose of squaring up all boxes on the mould conveyers, so that when they arrive at the main push-off position, they will be correctly situated on the mould conveyer, and can be pushed by the main pusher direct into the vibrating trough. It is pneumatically operated, both for

* From a Paper presented before the Institution of Mechanical Engineers by A. S. Beech, M.I.Mech.E.

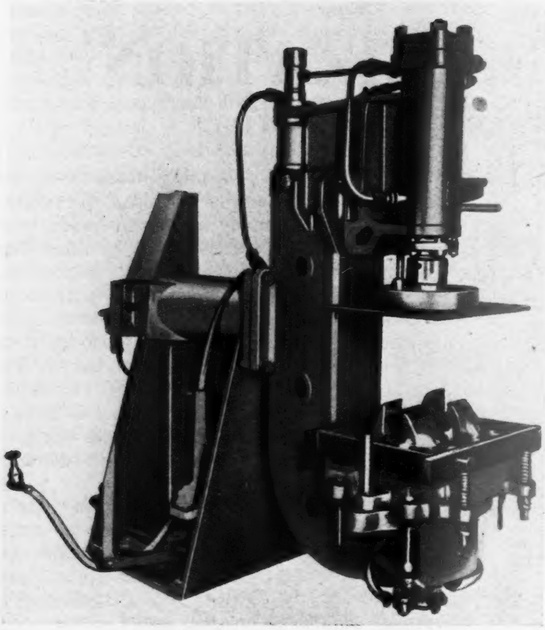


Fig. 2. Roll-over moulding machine.

together by the driven - roller conveyer.

When the moulding box, containing the castings and sand, arrives at the pre-pusher position, the pusher is automatically engaged and the box is located in a true position on the mould conveyor, being pushed right against the locating strip situated at the opposite side of the mould conveyer plate. Thus, all the boxes arrive in front of the main pusher exactly in the correct position for each pushing-off operation.

Immediately the boxes register their position in front of the pushing and return stroke. Dust and fume extraction plant is installed to keep all fumes away from the knocking-out station, and to rid the sand of the fine particles.

Operation. After the mould conveyer leaves the fume-extraction tunnel, the castings are sufficiently set to allow the boxes to be unclamped. The closing pins, however, are not removed from the boxes, as the box parts are not separated during the knocking-out operations; besides saving an operation, and the fact that the castings and sand are vibrated completely through the bottom box, it is much simpler to return the two half-box parts pinned

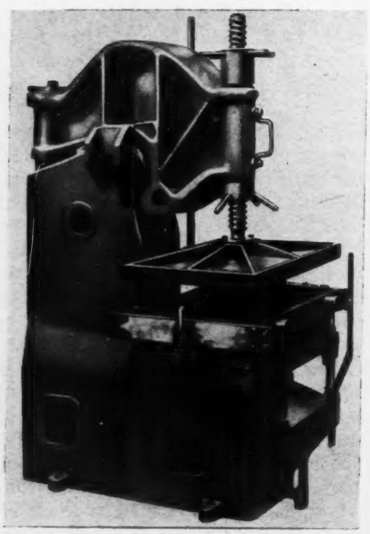


Fig. 3. Straight-draw moulding machine.

the knocking-out trough, the main pusher-valve comes into automatic operation, and the box is pushed from the mould conveyer on to the skid guides of the vibrating trough. At this point of the equipment, an extra precaution, in the form of a shear pin device, is taken to guard against any possibility of a break-down and a pile up.

As soon as the main pusher has projected the box, complete with sand and castings, on to the vibrating trough, the two box parts are supported from the top of the trough by means of the skid guides on the trough and the skid bars on the bottom half of the moulding box, and the closing pins that keep the two halves of the boxes together. The castings and sand are then vibrated through the open bottom part of the box, on to the grizzly bar base, and as the castings continue their journey along the trough, the sand falls through the grid on to a heat-resisting return-sand conveyer-belt, which conveys back the sand into the system for reconditioning.

The castings are then automatically discharged at the end of the trough on to the hot castings conveyer, which, in turn, feeds them into skips, which are attached to the pendulum cooling conveyer, and are passing at the head of the hot castings conveyer. Meanwhile, the moulding boxes continue their journey by vibration, assisted by gravity, and eventually are discharged on to the power-driven empty box return-track, which returns them to the back of the moulding machines. Fumes, steam, and dust are extracted by hoods over the vibrating trough, and also at the extraction points below the grizzly bar grids.

The entire operations of pushing the boxes off the conveyers, the knocking-out of sand and castings, and the return of sand to the sand plant and return of empty boxes to the machines, is carried out, even with the large quantities of castings to be produced, entirely by mechanical means and without the aid of one single man. Thus, there is complete mechanization.

Sand Preparation

Sand selection, sand conditioning, and sand reconditioning are of the greatest possible importance in the production of good castings, and the great parts that sand, and sand preparation and reconditioning play in the resulting castings are not always realized. It is not an exaggeration to say that at least 90 per cent. of scrap castings in the foundry are caused by sand and bad sand conditioning.

Sand Mill. The time-honoured mill formerly in use in most foundries was comparable to a mortar mill. Two heavy, narrow rollers rolled over the sand, which was contained in a stationary pan, and certain so-called up-to-date sand mills still retain this very objectionable feature. The full weight of these mullers passing over the top of stationary sand produces a crushing effect on the silica grains. Thus, whilst perhaps a good bond may be obtained, there is a considerable drop in permeability, and a large proportion of silt or dust is produced.

For moulding purposes the ideal state is that the sand shall be sufficiently permeable to allow the escape of the gases associated with the pouring operation, and its bonding qualities should be such that they will conform accurately to the shape of the casting, and withstand the wash of the metal. To obtain this ideal result the sand mill should be designed to give a rubbing effect on the sand, and not a grinding action.

The improvements incorporated in this mill may be summarized as follows:—

(1) The machine can be built to work either as a continuously operating unit, or as a batch unit. A lever lifts or lowers a ring fixed around the orifice for the sand discharge situated in the centre of the pan. If batch milling is required, the operator lowers the ring to prevent the evacuation of the sand, and raises the ring to automatically discharge the sand through the centre orifice. In continuous operation, the ring is kept open and the sand is fed through continuously, and at a predetermined rate by means of the diverters, which can be set for

angularity to mill the sand for a short or long period.

(2) The machine can be fitted with a cleaning plough, which will clean the pan surface of the machine, at the end of each shift, without the operator having to put his hands near any moving parts. To preserve the life of the pan, it is essential to see that sand does not stick to the surface, because sand left sticking to the surface becomes hard, and owing to the high silica content it eventually possesses almost a pumice effect, which will do great harm and cut into both pan and mullers very rapidly.

(3) The cleaning of the mullers in the old type of mill was carried out by long pieces of flat, high-tensile steel passing right across the muller surfaces. If, however, water was added more in one place than another, the sand at this particular part picked up more water and tended to become more sticky and to adhere to the surface of the muller, eventually cutting right into the scraper, which then had to be scrapped.

The new type of muller scraper is built up in sections, so that if uneven wear takes place only a small section will need replacement. In addition, the scraper is placed at a tangent in relation to the muller surface, instead of being flat against it, which tends to scrape the mullers much more efficiently.

(4) The pan, if desired, can be fitted with replaceable steel plate, so that when wear takes place, it is necessary to replace the plate only, and not the entire pan casting.

Roll-over Moulding Machine, Undersand Frame System. Fig. 2 shows a roll-over moulding machine with undersand frame. This type of machine, for deep or shallow work in large repetition quantities, is far superior to the jolt or jolt-squeeze type. The pattern is pushed into the sand, and not the sand into the pattern, and is drawn automatically on placing the machine to exhaust. This is certainly the most rapid and logical way of dealing with large quantities of repetition castings. The ramming is perfectly carried out, and all moulds are rammed up to the same density, provided the condition of the sand is controlled, particularly in respect to its moisture content. The bars in the boxes are solidly rammed underneath without any hand tucking, and the actual ramming of the mould is so quick that it is almost impossible to time it. The machines can be operated with either compressed air or hydraulic power.

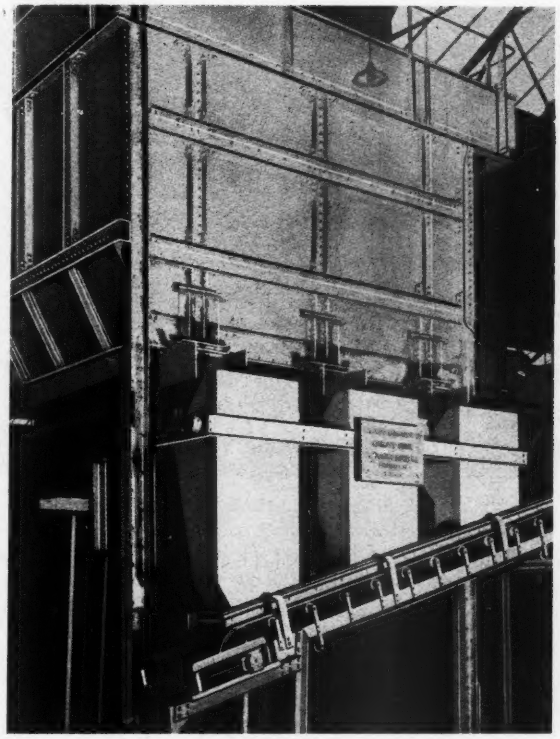


Fig. 4. Multiple-belt storage hopper.

Straight-draw Moulding. A machine for moulding the top parts of the moulds is shown in Fig. 3. As these do not need rolling over, the machine used is of the stationary straight-draw type. It embodies the same features as the roll-over model, the only difference being that as the mould has not to be rolled over, gravity is used for the pattern drawing. When the applied pressure is placed to exhaust, the pattern plate recedes through the under-sand frame, thus automatically drawing the pattern. On both types of machine the under-sand frames can be arranged to give the desired pressure to the mould.

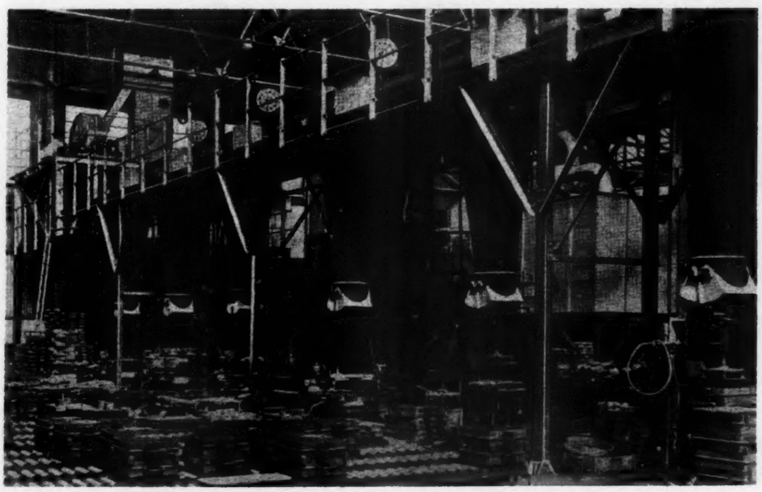


Fig. 5. Mechanized light foundry.

Multiple-belt Storage Hopper. One of the best recent improvements and one that has overcome a serious trouble in all mechanized or semi-mechanized foundry plants is the multiple-belt storage hopper (see Fig. 4). One of the bugbears of all mechanical foundry plants has been the sticking of sand to the walls of storage and feeding hoppers. This is particularly inconvenient when it occurs in the large storage hoppers, which are generally designed to stock quantities of sand ranging from 25 to 100 tons, or more.

The sand is generally hot and steamy when it enters the storage hopper, and consequently tends to stick to the sides of the hopper, whether the hopper is circular, with a revolving disc underneath, or rectangular, with feeder belt underneath. This results in the formation of "pipe" in the centre of the sand in the hopper, which tends to block the entire system. The sand has to be poked down with long rods, or the sides of the hoppers have to be hit with hammers, to dislodge the sand. The latter practice is very bad, because it dents the steel, and may aggravate

Fig. 6. Mechanized medium foundry.

the building-up tendency. This fault, which was invariably present in all storage hoppers, has been overcome by the hopper shown in Fig. 4.

Several belts, operated with push buttons, are situated across the narrow side of the bottom of the hopper. Three of such belts are shown in Fig. 4, but for large storage hoppers more would be introduced. When sticking occurs at the front or back of the hopper, the small individual belts are rotated by operating the separate push buttons, and the sand is fed into the general collector belt, which again feeds the sand into the system, and the storage hoppers are completely cleared, eliminating "sticking-up".

Mechanized Plant in a Light Foundry

The world renowned company manufacturing the famous Parsons turbines and electric generators, etc., have recently completely rebuilt and reorganized their three foundries (light, medium, and heavy), and have installed the very latest mechanized plant in each. Whilst their light foundry handles a certain amount of light repetition castings, the medium and heavy foundries' work is of a jobbing nature. The light foundry (Fig. 5) is mainly fitted up with sand conditioning, reconditioning, and distributing plants.

The work made on this plant is not of a really repetitive character because, whilst long runs of one casting occasionally occur, as a rule the patterns are changed very frequently, and the sizes of moulding boxes vary almost from hour to hour. Therefore, the mechanical mould conveyer has not been installed, but roller track is used.

Roller tracks for mould conveying in mass production defeat their own object. The moulds have to be pushed round by hand, and the waste of labour involved, which in the ordinary way would be undertaken by the mechanically operated conveyor, is entirely wrong. But in this instance it is the only way to deal with the problem. On the other hand, the plant is so designed that a mechanical mould conveyor can be installed at a later date, if the character of the work changes or more mass production in more regular sizes of moulding box is undertaken.

Mechanization in Jobbing Foundries

Medium Foundry. Many foundrymen throughout the world still adhere to the theory that mechanization methods can be applied only when the demand is for large numbers of castings off any given pattern, and they praise the efforts and results obtained in the production of repetition castings, but they still maintain that the jobbing side cannot be mechanized. As a result the jobbing shop, except in a few instances, lags behind. Certainly, mechanization or semi-mechanization of the jobbing shop is a much more difficult problem to solve, and will probably never be solved to the same extent as in the repetition shop, but a great deal can be done to lessen the laborious work in such shops which has been the rule in the past.

Fig. 6 indicates the layout for the manufacture of medium castings, ranging in weight from 1 to 12 cwts., all of a jobbing nature. It gives a very good general idea of what can be done to overcome the present world shortage of skilled moulders, and also to make foundries attractive to both the older and the younger generation. If it is possible to cut out the donkey work by the use of electric horse-power, and, at the same time, leave the skill, such as pattern drawing, gating, patching, etc., to the individual moulder, and make his surroundings more congenial, youths, who are so badly needed in the foundries, will be attracted, and the output per man employed will advance rapidly. Within six months of the installation of three mechanized plants at a works, twenty-five applications were received from youths for training. Previous to this there had not been one application for eighteen months.

The five jolt machines shown in Fig. 6 were already in the foundry before the other part of the installation was considered, and will probably be replaced by automatic mechanical sand-ramming machines. One such sand rammer is already installed on the mechanical moulding side of the

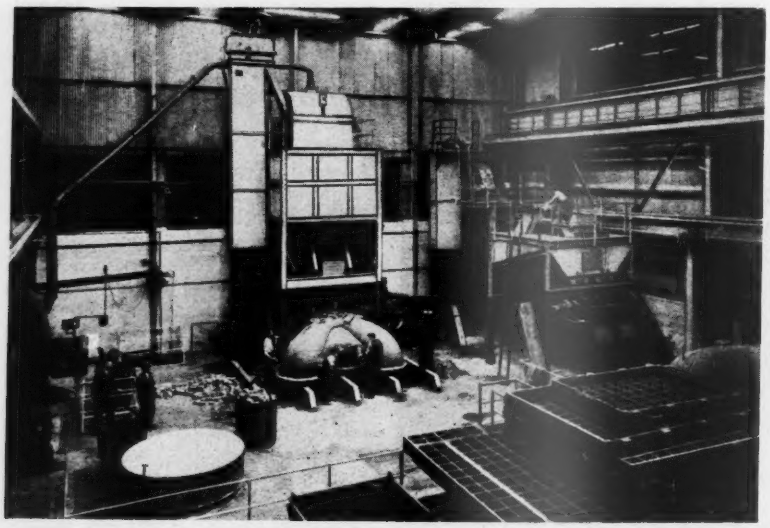


Fig. 7. Mechanized heavy foundry.

unit, and is much more elastic in its application than the jolting machines. At the present time, however, the moulds are rammed up on the jolt machines and by the sand rammer, and are transferred to the sand bed casting points by means of the overhead crane.

After pouring, and after the castings reach solidification, the moulds are knocked out in the vicinity of the knocking-out grid, and the sand then passes through the plant and eventually arrives in a re-prepared condition to the hoppers situated over the moulding machines, for use once more. All castings made on this plant are cast in green sand. If, however, dried sand moulds had to be used, portable mould dryers or stoves would be necessary, and the layout would be completely different. For such layouts as these, overhead cranes or mechanical lifting appliances are absolutely essential, as these are the only practical methods of transporting heavy moulds and castings of varying dimensions.

Heavy Foundry. Castings made in this foundry, Fig. 7, range in weight from 1 to 35 tons each. Some of the moulds are made of brick and loam, but in the main they are made of dried sand, are very heavily cored, and of a very intricate nature. All this sand has to be reconditioned, and as it includes large, hard lumps which have to be broken down before reconditioning, a special type of sand breaker has been installed.

There is a very large stationary knocking-out grid to deal with sand from the largest boxes. The other items in this plant are the magnetic separator, hexagonal rotary screen, 50-ton storage hopper, 15-tons-per-hr. continuous sand mill, disintegrator, and two 10-ton hoppers, from which finished sand in good moulding condition can be drawn. In addition there is a new sand hopper with a capacity of 10 tons, with all the necessary sand-feed belts, elevators, etc. The ramming up of these very large moulds is carried out mainly by the use of a tractor type of sand slinger of American design. The installation is served by powerful overhead cranes, and the entire plant gives every possible satisfaction.

Sand Crusher. This machine, see Fig. 8, is composed of an outside stationary cylinder, with a chute, into which the sand is introduced. Inside this cylinder is fitted a main shaft on to which are bolted a number of retaining discs, whose main function is to keep the loose crushing ball in between each disc from travelling laterally. The sand is introduced into the chute and as the discs rotate on the shaft the sand, in a lumpy condition, is directed forward to come into contact with the crushing balls, which reduce the lumps to grain size. The sand therefore travels forward, and is evacuated at the end of the drum, whence it is reintroduced into the sand system. Fig. 9 shows the inside view of the machine which operates very efficiently and effects much saving in sand.

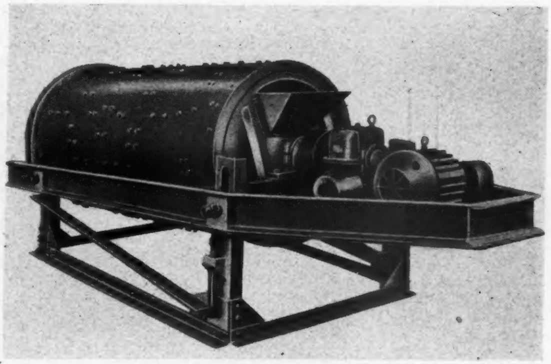


Fig. 8. Sand crusher.

Mechanical Sand Rammer. The type of machine shown in Fig. 10 is the British counterpart of a type that originated in the United States, and differs fundamentally and in details from it. The official name of the United States machine is the sand slinger, whereas, the British machine is called the sand rammer. One of the many points of difference is that it is believed in Great Britain that the machine should not be portable, but stationary, unless the portable machine possesses its own hopper or bunker from which it can obtain a supply of well prepared and conditioned moulding sand. Further, all foundries making fairly large castings, suitable for moulding with this machine, must possess overhead cranes, and, consequently, it is probably easier to take the job to the machine, than the machine to the job.

Moreover, the machine operates at such a speed that it has been found, from actual experience, that in order to obtain a uniformly rammed mould, it is essential to have a regular and continuous flow of sand to the head of the machine, and that this sand must be free of all "foreign" matter, particularly metallic inclusions. When the provisions possible in the portable machine are not present, a large packet of sand is fed into the head of the machine and is rammed into the mould, then for a few moments, very little or no sand arrives, during which time the impeller is thrashing air, until a further supply of sand arrives. Such conditions result in unevenly rammed moulds.

Apart from this, there is always a danger when taking a supply of sand from the foundry floor that a runner or git or some other piece of metal will be included in the sand, and as this will without doubt get into the impeller head, the result would probably be that some vital part of the machine would be smashed, and quite possibly the operator might be injured. The sand rammer or sand slinger is the only machine at present on the market which can successfully deal with jobbing work, or large castings which are too big for the average moulding machine.

Throughout the world hand moulders are scarce, and, consequently, wages are high. Apart from that aspect, however, the highly skilled man should not be used for the hard donkey work of shovelling sand into a moulding box, and the long and tedious work of ramming up large moulds. The skill of the hand moulder should be used to draw the pattern, or to do any patching to the mould, which is inevitable when a wooden pattern is used, as is usual in jobbing work. The skilled man should also supervise the core-setting and closing of the moulds, and generally finish the moulds ready for pouring. The machine should be left in the hands of an unskilled or semi-skilled man, who has only to guide the head over the box to solidly ram up the mould.

Thus, apart from the fact that this type of machine is a great time saver, it also is a great labour saver, and frees the skilled moulder for a much more important section of moulding, where his skill can be displayed to the utmost. It is so elastic that it is absolutely indispensable in any jobbing foundry

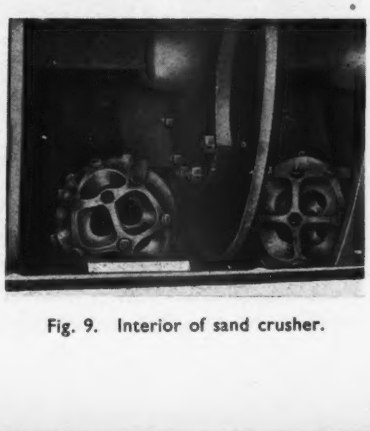


Fig. 9. Interior of sand crusher.

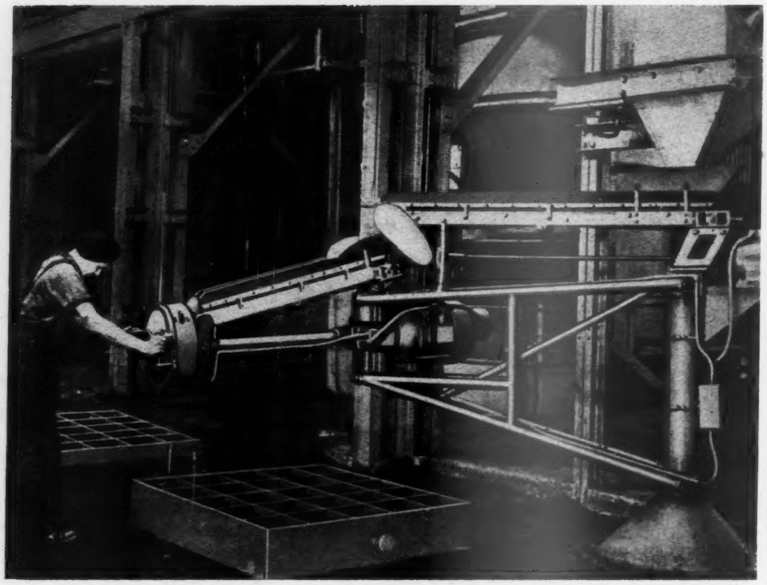


Fig. 10. Mechanical sand rammer.

or for special work, such as for standard items of large size castings.

Fig. 11 shows a fundamental difference between the United States sand slinger and the British sand rammer. The head of the United States machine is fitted with a drum carrying one cup or impeller revolving at a very high speed, which flings the sand into the mould at high velocity. It was found, however, that by reason of the shape of the cup the sand was thrown at a slight angle into the mould, and, moreover, that there was quite a considerable amount of spillage of loose sand into the moulds, which tended to result in uneven ramming.

Further experiments were therefore made in Britain, with a number of flat blades to replace the single cup, and a great improvement in the regularity of ramming was obtained, in fact, with the plural blades the ramming was quite uniform. It is also found possible to deal with a much larger quantity of sand with this machine than with the one with a single cup, and thus the capacity is greatly increased, without increasing the dimensions of the actual head.

Apart from the above advantages, the wear on the cup and on the liner of the United States sand slinger was very serious, owing to the abrasive action of the sand, the shape of the cup and its approach to the sand, which meant that the cups had to be replaced every eight hours, and the liners once a week.

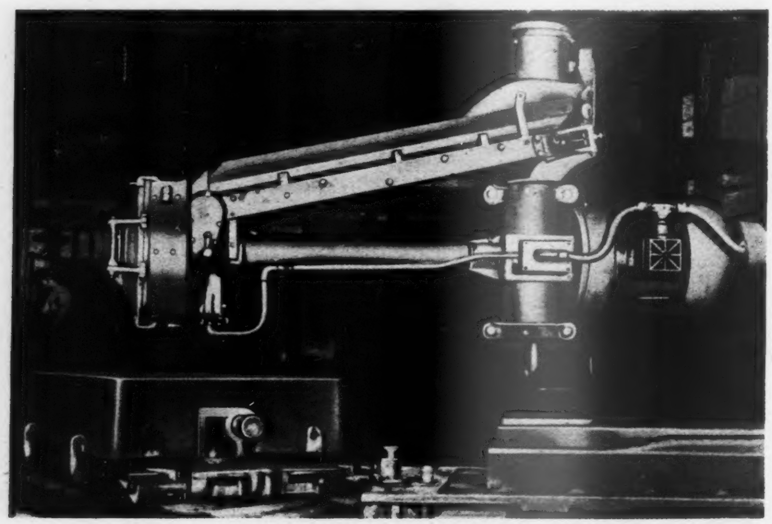


Fig. 12. Sand rammer installed in a large foundry.

With the British multiple-bladed type of sand rammer, the wear is very much less on both the blades and the liners and, as wear takes place on the blades, all that is necessary is to undo the screws, regrind the top of the blades, and replace them. The blades, which are made from ordinary mild steel, last for from two to three months, and the liners, which are made from high-tensile steel, last for about five to six weeks. The impeller drum in Fig. 11 has only two ramming blades, but the patent application covers a plurality of blades, as may be found necessary for larger sizes of head.

Sand Rammer for Large Works. For the repetition moulding of tractor frames and gearboxes, the size of the boxes are approximately 5ft. × 3ft. × 10in. deep, and the production obtained is ninety complete moulds in 8 hours. Fig. 12 shows a sand rammer installed in large motor works.

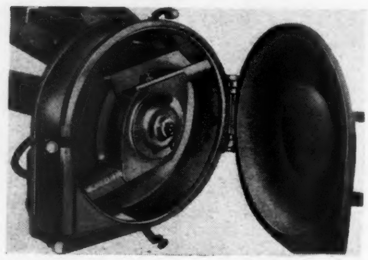


Fig. 11. Interior of the machine shown in Fig. 10.

Sand Rammer for Small Works. Fig. 13 shows the very latest innovation to the foundry. The small foundry had not been catered for up to the present time. Small foundries all over the world have had to make their living by making one or two castings off comparatively small patterns. Their only solution being to ram up such moulds by hand.

With the size and type of sand rammer shown in Fig. 13, small foundries will be able to utilize a mechanical ramming agent, and even for larger quantities of varying patterns, this machine can put them in a very good position to complete with the larger and better fitted up foundry. The machine has only just passed the design stage, and that shown in Fig. 13 is the prototype.

This type of machine must be fed with a continuous and regular supply of well-conditioned and clean moulding sand to procure the best results; part of it therefore comprises a feeding hopper, containing approximately four hundredweight of sand. The belts are controlled by push buttons from

the head of the machine, and, as the top belt is situated directly under the top hopper, as soon as it starts to rotate, a predetermined quantity of sand is fed down on to the lower gelt, and this in turn feeds into the head of the machine, which is fitted with a multiple-blade type of impeller to fling the sand at high velocity into the moulding box, thus ramming the sand securely and evenly around the pattern. The ramming arm swivels round at approximately 200 deg., and has also a backward and forward travel of 2ft. 6in. so that it can deal with all sizes of moulding box up to approximately 36 × 24 inches.

This machine has a very interesting field in regard to cores ; for very small cores probably the core blower is the most useful machine, provided that there are sufficient cores to be made of the same design and type, but the use of the core blower demands that core boxes must be specially constructed, and hence the core blower is only useful for core making when large repetition quantities are needed. The minor sand rammer, on the other hand, does not need special core boxes, and, consequently, it can ram up all cores up to 36 × 24 inches, whether for jobbing or repetition work. The minor rammer also shows great advantage over the core blower, for large-size cores, for either jobbing or repetition work.

Completely Mechanized Jobbing Shop

Mechanization can also be carried out in a jobbing factory. Consider a foundry doing jobbing work ranging from 1 lb. to 2½ tons in weight where some of the castings have to be made in green sand and quite a number of the larger ones in dry sand. There are therefore, three problems : small green sand work, medium green sand work and heavy dry sand work.

Each of these requires a different type of sand, and, consequently, three sand plants are installed. In the small green sand bay, a minor sand rammer running on a track, is installed, and is fed from the batch sand mill and plant. This machine goes up and down the track, ramming up moulds on either side of the track, and returning to the inclined belt feeder on the sand plant each time it needs a further supply of sand.

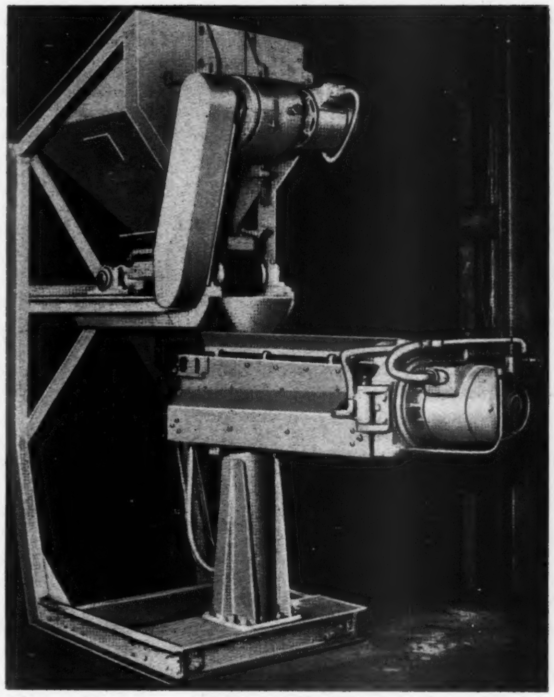


Fig. 13. Minor sand rammer.

The medium green sand moulds are made by the three "Linslade" sand rammers in the long green sand bay, and the sand is knocked out over the grids, after the moulds are cast, and returned to the plant for reconditioning and resupply to the various sand rammers at the moulding stations.

The dry sand work is moulded up by the two "Linslade" sand rammers, one of which is fitted with a 10-ft. arm, and the other with a 15-ft. arm, for very large work.

After moulding, the moulds are transferred to the drying stoves, and the dried moulds are then brought back to the pouring floor, for coring up, closing, and pouring. They are then again knocked out over the grids indicated, and the sand travels back to its own special plant for reconditioning and eventual return to the sand rammers for re-use.

No moulds are rammed by hand, and all the arduous and heavy work hitherto carried out by hand is entirely cut out.

Overhead cranes deal with the transportation of moulds, moulding boxes, metal, etc. The cores are rammed up by the minor sand rammer, and are then placed in the stoves for drying.

The metal is melted in three cupolas, each with a capacity of 3 tons of metal per hour, and a pulverized-fuel rotating-furnace is installed for the supply of any special mixtures of metal.

CORRESPONDENCE

Correspondence on subjects of technical interest is invited. The name and address of the writer must be given, though not necessarily for publication. No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter. If a reply by post be desired, a stamped addressed envelope should be enclosed.

DISC BRAKES

Sir,—I read with great interest your description of the Chrysler disc brake, which appeared very attractive on first sight. On closer scrutiny, however, it became evident that an apparently indifferent show brake—about two sizes too small—has been compared with a disc brake of adequate size, weighing at least 40 per cent. more and costing perhaps twice as much. The issue has been further obscured by the fact that whereas the disc brake was provided with an automatic adjuster, the shoe brake did not seem to have any.

No doubt, a disc brake has many advantages, particularly so far as heat expansion, rigidity and space considerations are concerned. On the other hand, whereas a modern sliding shoe brake can be "thrown together" the very rigidity of a disc brake seems to require much finer production limits. The virtue of rigidity is therefore somewhat double-edged. Space considerations may of course make a disc brake unavoidable in certain cases, but in other instances—especially on heavy vehicles—the available room in the wheel is essentially of cylindrical and not of annular shape. And with the trend towards smaller rim diameters, even on passenger cars, often the brakes have to spread laterally, favouring shoe brakes.

A perhaps not negligible point is that the presser plates seem to me inherently heavier and more difficult

to machine than ordinary brake shoes. I also suspect that in use, the ball-ramp cum sliding-dog arrangement is likely to develop a hysteresis effect. This may give rise to a noticably "wooden" feel; *i.e.*, once the brake is applied, its response to a slackening of pedal load may not be quite as instantaneous as that of a shoe brake. Further, automatic adjusters are not a privilege confined to disc brakes only. In fact, Hydrastatic actuation would seem rather difficult, although when well made it is known to impart a singularly "sweet" feel to shoe brakes.

A striking feature of the published experimental data was that the deceleration versus pedal load curves were convex, particularly so for the shoe brakes. In the best of circumstances this indicates an unpleasant feel, as explained on Fig. 1 of my article published in the same issue. This convexity may be of course due to heavy springs or linkage layout alone, but a "grabby" unstable brake usually exhibits the same characteristic too.

Finally, we might say that the quality of a brake installation is largely a matter of weight and costs. From this aspect there does not seem to be any evidence whatever to show that the disc brake described, offers any advantages. If anything, the opposite seems probable, because a 12" × 2⅝" shoe brake would provide the same lining and drum surfaces and would cost little more than the 12" × 2" brake tested. It may be also of interest that the two-leading-shoe brakes on British cars seem to stand greater HP/in^2 than the one quoted and fade is not one of their failings. Lest my remarks be misunderstood, I should like to say that they apply only to the particular case discussed.

G. A. G. FAZEKAS.

SIR,—The disc brake described in the "Automobile Engineer" of February 1951, page 69, is covered by the German patent 721.700 of 9.6.40, inventor Hermann Klaue, Dr.Eng., filed by the Argus Motoren Gesellschaft m.b.H., Berlin-Reineckendorf and the cognate French and Italian patents No. 872.627 and No. 391.734.

Dr. Klaue has published a detailed description of this brake in the "Automobiltechnische Zeitschrift" 1942, No. 18. The disc brake has met with practical regular application during the war in its design with compressed air operation in traction machines for the armed forces as a steering and travel brake and in its design with mechanical operation as a brake for tanks (e.g., Tiger and Panther).

After the war this brake was further developed by the Klaue - Bremse G.m.b.H., Ueberlingen/Lake Constance for peace-time purposes. In the meantime, the series production of bicycle and motor cycle brakes has been started up, while the compressed air and hydraulic disc brakes will be appearing on the market shortly.

DR. ING. HERMANN KLAUE
Überlingen/Bodensee
Germany.

POLAR MOMENT OF INERTIA

SIR,—In the interesting article in your last number entitled "Future Tendencies" you say:—

'Since the engine output is expected to exceed 300 h.p., the car is obviously capable of extremely high speeds and in these circumstances a high polar moment of inertia seems to be desirable'. If by this is meant the polar moment of inertia about a vertical axis through the centre of gravity of the car, surely it is a low not a high polar moment of inertia which should be aimed at.

In an article published in the *Automobile Engineer* of January 1930, I analysed at some length the motions which take place during steering and skidding. Without quoting the details of the calculations, it is, I think, clear that the final limit of steering control of a car is set by the point at which the front wheels begin to slip sideways during any agreed rate of rotation of the steering wheel, such a rate being that necessary for conveniently quick alteration of the car's direction.

For any given weight on the front axle and any given co-efficient of friction of the tyres, the resistance to alteration of the car's direction depends on two factors—the polar moment of inertia about a vertical axis through the centre of gravity of the car, and the angular acceleration about the same axis, this in turn, depending on the speed of the car and the rate of angular movement of the steering wheels about their swivel pins.

Therefore the higher the speed of the car, the lower should be the polar moment of inertia, if an agreed rate of manipulation of the steering is to be maintained. Without going to the limit of front wheel adhesion, steering tends to be made unpleasant by a relatively high moment of inertia. The experiment of driving a small car with a long ladder lashed on board shows this.

Yours faithfully,
G. SARTORIS,
A.B.C. Motors Ltd.

(While theoretical analysis may show that a low polar moment of inertia appears desirable, the evidence from practical experience with high-speed cars does not seem to support this view. The following quotations are from B.I.O.S. Final Report No. 1755 "Investigation into the Development of German Grand Prix Racing Cars between 1934 and 1939". The views of the Mercedes technical department are summed up thus:—

"The primary objection raised to rear engine mounting was on the grounds that this arrangement normally results in the greater part of the total weight of the machine being concentrated about its central vertical axis. This is very undesirable as it reduces the polar moment of inertia of the car to a relatively low figure. This in turn detracts from the straight running tendency of the car as any acceleration about the vertical axis of the car due to any external influence will develop at a relatively high rate in consequence. . . .

There can be little doubt that the above disadvantages assumed serious proportions in practice and were largely responsible for the inferior road holding of the Auto-Union to that of the Mercedes machines on fast sinuous circuits".

Referring now to the layout of the Auto-Union car the following quotation is taken from British Patent Specification 425937 filed by Dr. Porsche in respect of his rear engined chassis layout as later applied to the Auto-Union machines:—

"Through the main masses, that is to say the relatively heaviest sprung masses being grouped about the gross centre of gravity of the sprung masses . . . it is sought to prevent any turning of the car about the vertical axis passing through the centre of gravity, even with variations in the adhesions of the wheels, owing to the inertia forces acting on the chassis. In this case, the chassis, owing to the loads acting on it being grouped centrally, has such a small moment of inertia about this axis that the dangerous slewing of the car can be prevented owing to the absence of an effective slewing moment".

Referring to the Mercedes statement that rear engine mounting was unsatisfactory for racing cars, an Auto-Union representative "agreed that the objections raised on the grounds of the inherent low polar moment of inertia and forward driving position were valid . . . He also agreed, however, that constant weight distribution (with varying fuel load) combined with a high polar moment of inertia about the central vertical axis was the ideal to be aimed at and that this might finally be

achieved more easily with a forward engined chassis arrangement".

The compiler of the Report makes the following comment on this aspect of chassis layout:—

"With reference to the reduction of the slewing of the machine about the central vertical axis, it can be seen that the reasoning of Dr. Porsche on this matter was directly opposed to that of the Mercedes engineers, the former seeking to reduce the slewing moment and the latter to modify its effect. Dr. Porsche makes the assumption, however, that the forces tending to slew the chassis about the vertical axis passing through its centre of gravity are proportional to the distance of the individual centres of gravity of the main masses from the said axis. It has apparently not been recognised that under certain conditions when cornering, the deflecting forces acting through the wheel centres due to centrifugal force at a given instant are a function of the speed, total weight and the cornering power of the tyres alone. Under these conditions the above assumption does not hold good and the greatest possible polar moment of inertia about the vertical axis is desirable to limit the rate of slewing acceleration, to the lowest attainable figures".

Apart from these aspects of the question, aerodynamic stability may have considerable effect. With a well streamlined body it appears that at small angles of attack the centre of pressure may be very far forward, thereby introducing a considerable moment tending to deflect the car from its course. This also might introduce handling difficulties with a car having a low polar moment of inertia.

In the circumstances, it is considered that more information is required on the whole subject, but the weight of the available evidence is in favour of a high polar moment of inertia.—*Ed.*).

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

The Motor Vehicle

By K. Newton, M.C., B.Sc., A.C.G.I., A.M.Inst.C.E., M.I.Mech.E., and W. Steeds, O.B.E., B.Sc., A.C.G.I., M.I.Mech.E.

LONDON: ILIFFE & SONS LTD., Dorset House, Stamford St., S.E.1. 1950. 590 pp. $8\frac{1}{2} \times 5\frac{1}{2}$. Price 35s.

There are to-day a large number of books describing the mechanics of the motor vehicle from which the student, the motor vehicle designer, or the vehicle operator may make a choice. Many of these are less expensive than this volume. If, therefore, as in the present instance, a book on this subject runs to four editions and maintains its sales so high that nine impressions of the last edition were required, it must provide value for money.

Beginning with the fundamentals of automobile engineering, this section covers engineering drawing, bearings, gearing, and principles of mechanics. The book is divided into two parts, The Engine, and Transmission respectively. After discussing the general principles of heat engines, and engine balance, the authors consider the structural details of the engine. Under this heading mention is made of various types of pistons and rings, bearing metals, built-up crankshafts, valves and valve inserts, camshaft drives, hydraulic tappets, chain drives, etc. This chapter concludes with descriptions of Dennis, Lea Francis, and Vanguard engines in order to show clearly the relationship of the various components in the general engine arrangement.

A chapter is devoted to the compression ignition engine, much of this describing the various types of injection equipment. Representative examples of c.i. engines considered in detail are the Gardner LW the Perkins Panther, the Morris Commercial $4\frac{1}{4}$-litre, and the Albion 4-cylinder engines. In two stroke engines, the Trojan, General Motors, and Foden engines are described. The following chapter is on carburation, and possibly a more logical sequence would be to have considered engine parts before engine assemblies.

Practically every type of carburettor in use is described and illustrated in line drawings. Chapters on fuel supply including air cleaning and supercharging. Lubrication, cooling and future trends, complete the first section. In considering future trends, the possibilities of petrol injection and the gas turbine are studied.

In the section on transmission, much new material is included on automatic transmissions, torque converters, synchromesh mechanisms, rebound dampers and power steering. Revisions have also been made in the chapters dealing with brakes, gear boxes, and independent suspensions. Many new diagrams have been added. In fact, the book is as much up-to-date as it could be. To make room for the additional matter, the chapter on ignition equipment has been omitted, since as the authors state in their preface, many books now are available which deal exclusively with this subject.

Carburettors and Fuel Systems

By Arthur W. Judge, A.R.C.Sc., D.I.C., Wh.Sc., A.M.I.Mech.E.

London: CHAPMAN & HALL LTD., 37, Essex Street, W.C.2. 1950. 505 pp. $4\frac{3}{4} \times 7\frac{1}{4}$. Price 12s. 6d.

This is the fifth edition of Volume II of a series of Motor Manuals by the same author. It is essentially the same as the last edition published in 1941, excepting that a new chapter of some sixty pages has been added giving details of recent developments. These later developments include improved anti-knock fuels, modified acetylene fuels, butane, methane, alternative fuels, the Solex dual-port downdraught carburettor, the Zenith type V.M. carburettor, the Stromberg type DBV carburettor, Amal carburettors, triple S.U. carburettors, the Thornycroft petrol injection engine, and many notes on matters relating to carburation and fuel.

Introductory chapters cover the fundamentals of the subject, and any garage mechanic should be capable of understanding these and profiting from them. Almost every type of carburettor is described and illustrated, and the illustrations are valuable. Separate chapters are devoted to American carburettors, motor-cycle carburettors, and paraffin and other carburettors. Again, the sectional diagrams are just what the reader will require. Of interest also is the chapter on floatless carburettors, giving illustrations of those designed for model engines.

Under "Vaporising the Mixture" will be found details of methods of heating the air intake, hot water jacketing, hot spot methods, regulating exhaust heating, vaporizers of many types, and auxiliary starting devices. Fuel feed systems are comprehensively described, and the author fully realises that the design of inlet manifolds has considerable bearing upon carburation. The notes on testing, tuning and trouble tracing include details of the equipment normally employed and recommended for use.

The book covers carburettors and fuel systems not in use to-day, those in use but not now fitted, and the modern types of carburettors, fuel pumps and gauges.

The Motor Trade

By H. E. Milburn, M.I.Mech.E.

LONDON: *Published for the Motor Trader by The Trader Publishing Co. Ltd., and distributed by* ILIFFE & SONS LTD., Dorset House, Stamford St., S.E.1. 1950. 345 pp. $5\frac{1}{2} \times 8\frac{1}{2}$. Price 21s.

Anybody contemplating entering the motor business, or thinking of running a garage, those who have little experience of this work, and even those who have been long engaged in motor selling and servicing, will all find this book of value. Entering any business to-day presents a complex matter, and the motor trade has special difficulties. The term "motor trade" may apply to mechanical and body repairs, electrical servicing, buying and selling second-hand vehicles, private hire, driving tuition, petrol and oil sales, battery charging, hire purchase on new cars, etc. A guide to many of the problems encountered in these various branches are all to be found in this book.

That opportunities exist in the motor trade for ambitious people is not disputed by the author but he also suggests many methods of encouragement and expansion for those already engaged in this business. Suitably enough, his first chapter is Starting From Scratch and gives advice to those starting a new business. He covers town planning bye-laws, capital, partnerships, limited liability, buying a business, etc.

The chapter on planning the premises,

followed by lighting, repair shop equipment, stores, etc. give sound information to everyone contemplating new garages or extending their present ones. The author also has considerable advice to give concerning vehicle selling, repairs and service, from a sales point of view.

The notes on workshop control and management, office organization and accounting, costing, customer records, engagement and payment of employees, insurance, etc. are all subjects that should be read by the most knowledgeable motor trader. In parts, the information may be slightly out of date as regards salaries payable since these are rising at the present time, but it is difficult to criticize other facts given in this book.

The chapter on driving tuition and private hire will be useful to those engaged in this type of business. Notes are given to help the operator to arrive at fees and charges that cover depreciation and other expenses. There is a chapter on advertising and publicity, which is a subject on which most garage owners, large and small, are more or less ignorant. Finally, a list of the trade associations and organisations, with their rules for membership, their objects and other relevant information is provided.

As the author remarks in his introduction "The trade has a great fascination for many not engaged in it as well as for those who live by it. There is a great satisfaction in giving useful service to a public which is frequently unreasonably exacting in its demands". It is wished that many of the general motoring public would read this book. It is certain that a great number of motor traders will do so.

Motor Oils and Engine Lubrication

By Carl W. Georgi.

New York: REINHOLD PUBLISHING CORPORATION. London: CHAPMAN & HALL LTD., 37 Essex Street, London, W.C.2. 1950. 514 pp. Price 68s.

It is generally acknowledged that the U.S.A. are ahead of us in matters relating to lubrication. Oils of the additive type for heavy duty, detergent grades, etc., all originated in the United States and were quite widely used there before being introduced to this country. It is not surprising, therefore, that what is believed to be the first book dealing solely with automobile engine lubrication should be by an American author.

The author is Technical Director of the Research Laboratories at one of the leading American lubricating oil companies and is Vice-President of another. He has contributed innumerable papers on the subject of lubrication to many engineering institutions and societies. The subject is one of interest to every automobile engineer, and this book, in spite of its high cost, should be available for reference to all of them.

Beginning with classifications, definitions and specifications, it is unfortunate that the book has just appeared before the revised figures were available for the S.A.E. system of classification for crankcase oils, appearing on the first page. If readers correct these, there appears to be no other material in this very complete volume that is out of date, or will be for some time. Viscosity and viscosity index are dealt with very thoroughly and related to such matters as oil consumption, cold starting, and power loss. The author is also aware of the importance of viscosity-pressure relationships.

Details of engine tests in use in America and coming into use elsewhere are given, with illustrations of parts from engines that have been lubricated with differing grades of lubricants. The next chapters are devoted to refining and to additives. The chapter on synthetic oils deals with those made by Fischer-Tropsch synthesis, the "Voltol" oils developed in Belgium, those developed in Germany, and Japanese synthetic oils.

The remainder of the book is practical. Beginning with Engine Lubrication, costs are analyzed, conditions met with are discussed, and engine oil failures are touched upon. The effects of engine trends on lubrication are covered, and the vexed question of when to change motor oil is discussed sensibly and at length, in relation to all types of operating conditions. Oil consumption is covered in its own chapter, and both users of vehicles and chemists will be interested in the chapter on Sludge and Deposits. In this chapter the author has much to say about detergent oils and why they do, and sometimes do not, eliminate deposits. His discussion is frank and unbiassed.

It is satisfactory to note that a chapter on crankcase ventilation has been included, and diagrams of all available systems are given, together with illustrations of parts damaged by poor ventilating systems. The effect of fuels on lubricating oils are considered somewhat briefly in this otherwise very detailed and fully comprehensive volume, but the ill effects of sulphur in fuel are noted.

All engine parts that may suffer from faulty lubrication are discussed in detail, including bearings, valves, pistons and rings, spark plugs, and oil filters. Only a few pages are devoted exclusively to diesel engines, although the differences between these engines and petrol engines are given, and there is probably no need to discuss them further by themselves. Further mention of these engines is of course made when dealing with Heavy-Duty oils.

This is a valuable book, and will probably be considered the standard work on automobile engine lubrication. In conjunction with its very long lists of references appended to every chapter and covering published material from all over the world, the reader should have no difficulty in pursuing any particular aspect. Half tone and line illustrations, graphs and charts appear in all cases where they amplify and simplify the text.

Mechanical World Year Book, 1951

MANCHESTER: EMMOTT & CO. LTD., 51, King Street West. 1951. 4 × 6¼. 508 pp. including manufacturer's announcements. Price 3s. 6d.

This popular year book, a companion to the Electrical Year Book, enters its sixty-fourth year of publication and retains its established format and material. New to this edition, however, are twenty-five pages devoted to Engineering Aspects of Productivity, written by Dr. Galloway of the Production Engineering Research Association. In this new section, the importance of correct selection of materials and the correct use of machine tools is discussed. The author has much to say about surface finish and the methods of achieving it. The importance of correct cutting fluids is also emphasised, a subject that has been under review by P.E.R.A. recently.

The rest of the book follows usual practice, and some revisions have been effected. In addition to many other sections, the following are perhaps of special interest: Continuous Flow Gas Turbines, Fuel and Lubricating Oils, Machine Tools, Diamond Tools and their Applications, Light Alloys, Die-Casting, Plastics, Steam Boilers, Mechanical Press Work, Production of Gears, Steam Turbine Notes, and very many tables and charts such as are in daily need by practical mechanical engineers. The book continues to provide extremely good value for money.

Mechanical World Electrical Year Book, 1951.

MANCHESTER: EMMOTT & CO., LTD., 31, King Street West. 1951. 357 pp. 4 × 6¼. Price 3s.

In its forty-fourth year of publication, this pocket book for electrical engineers still retains its popularity. The amount of information provided, most of it of daily use to the pratical electrical engineer, and even to the mechanical engineer, is amazing value bearing in mind the low price of the book.

There have been many minor revisions and an additional section is now included on Space Heating. This deals with high-temperature and low-temperature heating giving the coefficients of heat loss for various types of construction for various types of rooms, etc. Heaters discussed include tubular, convectors, unit air heaters, and radiators. Water heating systems include immersion heaters, circulators, self-contained water heaters, and electrode boilers. The information is, of course, brief, but the facts and figures provided are those to which the practical man may often wish to refer during the course of his work.

Subjects which again reappear include amongst many others, Electrical Calculations, Turbo-Alternators, Control Gear, Switches, Automatic Protection, Regulations for Electrical Equipment of Buildings, Electric Welding, and numerous tables and charts of everyday use for reference purposes.

Motor Cycle Cavalcade.

By Ixion of *The Motor Cycle.*

London: ILIFFE AND SONS, LTD., Dorset House, Stamford Street, S.E.1. 1951. 8¾ × 5½. 272 pp. Price 10s. 6d.

Although essentially popular in character, this is a book that would interest most people in the industry. For those associated with motor cycle work, it will of course have a special appeal. Written by Ixion of our associated journal, the *Motor Cycle*, it provides a highly entertaining account of the days that are gone, as well as a present survey, and forecast of things to come. Commencing with the experiences of a pioneer rider and descriptions of the machines of that period, the story is taken in detail to the present day. There is a considerable amount of technical detail, particularly as this relates to the gradual evolution of those features that constitute the more or less standard layouts of to-day.

A brief history of all the various makes is given, and one of the most interesting features of the book is a photogravure section of over 100 illustrations, many of which are of definite historic interest. Other photogravure sections give pictures of notable personalities, events and machines. Every aspect of the subject is dealt with, including trials, racing, records, etc. In short, the interest is sustained from cover to cover.

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

AN improved type of Coventry self-opening diehead has been developed by Alfred Herbert Ltd., Coventry. It is designated style C.H.S., and incorporates design features that have contributed for so long to the success of the style C.H. Coventry diehead. At present, $\frac{1}{4}$in., $\frac{3}{8}$in. and $\frac{1}{2}$in. sizes are available for early delivery. The $\frac{3}{8}$in. C.H.S. diehead replaces the $\frac{5}{16}$in. C.H. diehead. Each size is made in four types which are illustrated in Fig. 1.

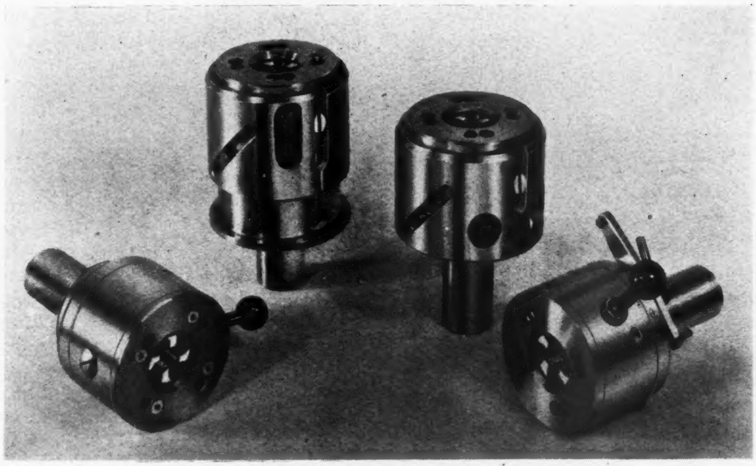


Fig. 1. C.H.S. Coventry self-opening dieheads.

The various types and their uses are:

Standard Stationary Type for use in capstan and combination turret lathes and single-spindle automatics of the Brown and Sharpe, B.S.A., C.V.A., Index, Herbert, Butterworth, Cleveland and Footburt types where the diehead does not rotate. This diehead is opened by arresting the travel of the turret or slide carrying it. It is closed by hand.

Standard Rotating Type with a closing sleeve and shank. It is for use on drilling machines and single-spindle automatics such as Petermann, Wickman, Brown and Ward and Bechler on which the diehead rotates. Special shanks or adaptors can be supplied when necessary. Opening of this diehead is effected by arresting the travel of the spindle. It is closed by means of a closing bracket or yoke.

Rotating Diehead with closing sleeve and detent withdraw sleeve, for use on multi-spindle automatics of the Acme, Gridley, New Britain-Gridley, Conomatic and Cleveland types. This type is opened and closed by means of a yoke or closing bracket.

Stationary Diehead with detent lever attachment for opening. This may be used for work as short as three threads in length and for operations on which the diehead is leadscrew controlled. Opening is effected either by operating the detent lever by hand or by some form of stop, such as a tool in the rear tool post of a capstan lathe.

The standard stationary, the standard rotating and the rotating dieheads are fitted with an improved type of spring shank which permits threads up to the full capacity of the diehead to pass clean through the shank, so that thieads of any length may be cut. This type of shank enables a good start of thread to be obtained by semi-skilled operators. It also makes it easier to pick up a second cut when roughing and finishing cuts are necessary, and it facilitates the production of fine threads on heavy turret lathes. The detent pin which controls the opening of the diehead can be quickly removed for regrinding or replacement.

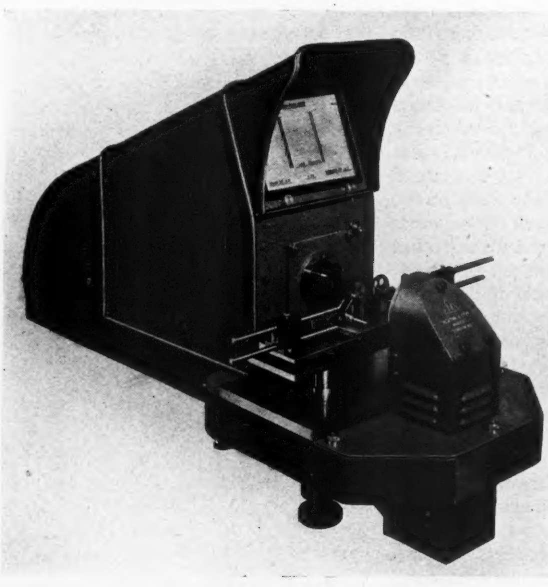


Fig. 2. Newton projector.

All the stationary dieheads are fitted with roughing and finishing arrangements to allow two cuts to be taken without any alteration of the initial setting. This produces a better finish on the thread, particularly on tough materials. Threads slightly larger in diameter than the nominal capacity of the diehead can be screwed, but these are limited in length and must be of fine pitch. Taper threads with a maximum length slightly less than the width of the dies can also be cut. By using left-hand dies, left-hand threads can be cut as easily as right-hand. Provision is made for fine regulation of diameter, and an efficient lock prevents movement under working conditions.

With the exception of the $\frac{3}{8}$in. diehead, which is a new size, the dies used are exactly the same as those for the equivalent sizes of the C.H. diehead, and they can be ground in the same die grinding fixture. The dies for the $\frac{3}{8}$in. C.H.S. diehead are of the same section as those for the $\frac{1}{2}$in. C.H. diehead, but are shorter. The die grinding fixture for the $\frac{1}{2}$in. C.H. diehead has been modified to allow dies for the $\frac{3}{8}$in. C.H.S. diehead to be ground in the same die grinding fixture.

Profile Projection

An interesting profile projector suitable for use on the shop floor for checking large quantities of small parts

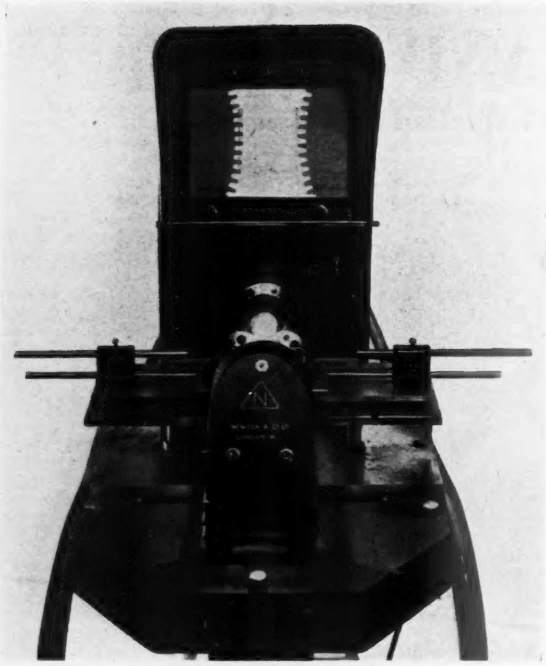


Fig. 3. Newton projector set up for inspecting small gears.

has recently been developed by Newton and Co. Ltd., 72, Wigmore St., London, W.1. There are two models. The larger, shown in Figs. 2 and 3, has a magnification of ×25. It is claimed that at this magnification the stage error does not exceed 0.001in. over a screen area of 15in. diameter. The smaller instrument gives a ×10 magnification. Respective screen dimensions are 12in. × 10in. and 6½in. × 4¾in.

The projector chassis is of substantial box section. It is mounted upon a stand of welded, tubular construction. Three mounting points are provided at the base of both the chassis and the stand. Work illumination is by means of a 12V 96W coiled-coil filament type bulb in a cast aluminium lamphouse mounted at the front of the instrument. Adjustment of the bulb in a vertical plane is effected by the relative movement of two knurled nuts. These nuts determine the position of the bulbholder supporting pillar and they allow a fine degree of adjustment to be obtained.

A conventional optical system is employed. The light beam is condensed by two lenses and is reflected through the objective bya mirror mounted at 45 deg. in the top of the lamphouse. To allow the light beam to be centred accurately, the mirror is attached at three points, each of which is adjustable. The mounting of the condenser lens at the front of the lamphouse is detachable. It can be removed for cleaning or for changing lenses.

Two accurately ground steel plates, one mounted upon the other, comprise the worktable. The upper plate is free to move in a direction parallel to the axis of the lightbeam for focusing the work. It has three hardened, ball-ended pins that protrude from the underside. The lower plate is slotted to accept a clamp rod and has on its upper surface a 90deg. vee that locates, in a lateral plane, two of the three pins that protrude from the upper plate.

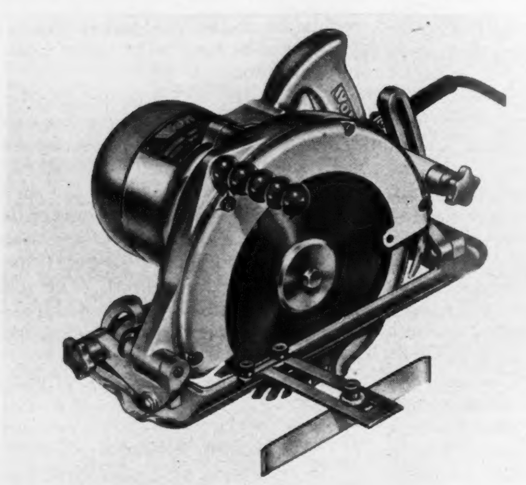


Fig. 4. Wolf R.S.10 electric saw.

At the rear of the projector there is an aluminized mirror that reflects the image on to the objective screen. It is mounted on three adjustable spring-loaded screws and is adequately protected by a reasonably dust-proof cowling. Two sheets of glass between which a lay-out drawing can be placed compose the object-screen. Two small independently controlled cams upon which the lower edge of the screen rests, give a degree of adjustment for lining-up the diagram with the silhouette of the workpiece.

As these projectors have been specifically designed for quantity inspection the development of suitable work fixtures is important. In a typical example for checking small gear wheels, the wheels are mounted on their spindles in two vee blocks and the profiles of the teeth projected on to the screen. The vee blocks are free to slide at right angles to the axis of the light beam and each can be moved independently to relate the projected profile of the wheels with each other and with tolerance bands on the screen. When the desired position is obtained, the blocks are clamped in position.

This set-up gives a rapid means for checking the concentricity of the periphery and for determining the consistency of the tooth depth. When the wheels are spun about their axes, the toothed portion appears as a half-tone on the screen and any irregularity can readily be seen. The magnitude and direction of the error can be measured by slowly rotating the gear and comparing the image with the tolerance bands. If necessary, the profiles of individual teeth may be checked by substituting a template of the tooth form in place of the tolerance band chart.

When it is desired to check small turned shafts that are too long to be projected completely upon the screen, a sliding fixture may be used in conjunction with the end-stops fitted to the worktable. A component of known dimension is clamped in the

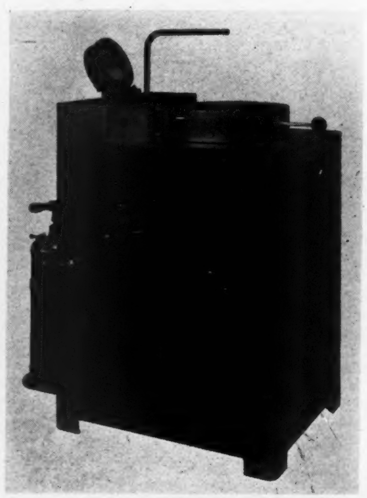


Fig. 5. Wild-Barfield VWPI general purpose furnace.

fixture and the stops on the table are so set that upon sliding the fixture first to one stop and then to the other, the silhouette of the ends of the shaft bear a correct relationship to the lay-out on the screen. To facilitate the production of accurate lay-outs a special drawing board can be supplied. It is accurately scaled to factors of ×25 and ×10.

The projector can also be used to inspect the surface of metal parts and for crack detection. Through the use of an adjustable lamphouse and the attachment of a small mirror to the lens, an intense beam of light may be directed along the optical axis of the instrument and reflected by the metal surface back to the lens for examination of the work.

Electric Saw

A recent addition to the range of tools manufactured by Wolf Electric Tools Ltd., Hanger Lane, Ealing, London, W.5, is illustrated in Fig. 4. It is a 10-in. portable, all ball-bearing electric saw, type R.S.10, that has been developed to meet the demand for an electric saw with larger power reserves, lighter weight and greater reliability than hitherto available. The use of magnesium alloy in its frame construction has not only effected important savings in weight, but it is claimed has also made possible the production of a tool with a greatly better performance than any similar tool. An extremely powerful motor of advanced design is incorporated and special care has been taken to ensure perfect commutation and cool running under maintained heavy load.

A spring-loaded safety guard of sheet steel completely covers the blade. It is pushed back as the machine is fed in to the work, and provides the operator with complete protection under all conditions. The actuating mechanism for the guard is in a dust and swarf-proof housing, while the heavy sheet steel sole plate, in common with all other steel fitments, is cadmium plated to give protection against corrosion.

To ensure accurate positioning for straight and bevel cuts up to 45deg., a line cut indicator is provided on the leading edge of the sole plate. The depth of cut is controlled by handwheel adjustment on a vertical quadrant, and the addition of a riving knife, made of tempered steel, effectively relieves back pressure on the saw blade.

A useful range of blades is available. The blades are all of crucible steel and the range is adequate for all general purposes. For ripping and cross-cut work there is the combination blade which is standard equipment on all machines, and for a smoother cut there is the cross-cut blade. For cutting wall-boarding, composition and the like materials, there is a fine tooth blade known as the compo and wallboard blade. It is specially tempered for use where an occasional nail may be encountered in cutting. The friction blade is suitable for cutting corrugated metal sheet, and the planer blade for plywood. A ripping guide is provided in the standard equipment. It can be adjusted for ripping or crosscut work up to 5½in. in width and is readily attached or removed.

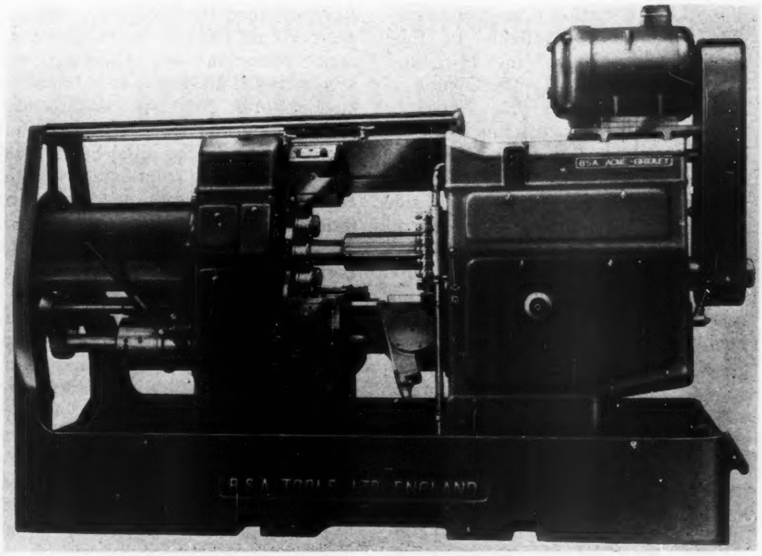


Fig. 6. B.S.A. Acme-Grindley six-spindle bar automatic.

General Purpose Furnace

To bridge the gap between the smaller light duty furnaces and the heavy duty full-scale production furnaces, Wild Barfield Electric Furnaces Ltd., Elecfurn Works, Watford Bypass, Watford, Herts., have recently developed a general purpose vertical furnace that is a self-contained unit capable of operating at a chamber temperature of 1050 deg. C. with safety. It is made in two models. One, designated VW1 is a straightforward furnace without any form of atmosphere control. The other, the VWP1, which is illustrated in Fig. 5, is identical in construction but has the added advantage of the Paragen Burner system of atmosphere control. Each type is designed to deal with components of slender section or with batches of components that can be suspended in jigs and quenched as a complete charge.

The heating chamber, which is 10in. diameter and 20in. deep, is constructed of high grade moulded refractory sections built to form a cylinder having on its internal surface a groove to carry a heavy nickel-chromium helix. As the element is free to radiate heat within the chamber, it operates at a lower temperature than elements placed outside the refractory. This ensures a longer element life for a set furnace temperature. Split type doors are fitted. When closed they effect a seal to the chamber and so minimise heat losses. They also allow the suspension of work singly or in batches on jigs from the tool post mounted at the back.

An indicating pyrometer is mounted on top of the furnace. The energy regulator control panel, with isolator, the necessary fuses, contactor and control circuit terminals, is built into the framework of the furnace at the rear. To safeguard the operator and the equipment, door switch and excess temperature cut-out protective devices are fitted. If desired, fully automatic temperature control can be provided.

The Paragen atmosphere control unit fitted to the VWP1 models considerably increases the utility of the furnace. It is simple and economical in use and provides an atmosphere that gives practical freedom from decarburisation to a wide range of engineering steels during heat treatment. The atmosphere is produced from paraffin or kerosene, a cheap, readily available commodity. The plant itself comprises three main portions, a fuel container, a pump and pressure gauge that are combined with the container, and a burner for vaporising the fuel.

Six-spindle Automatic Bar Machines

B.S.A. Acme-Gridley six-spindle automatic bar machines are now being manufactured under licence from America by B.S.A. Tools Ltd., Macka-

down Lane, Marston Green, Birmingham. They are available in four sizes, namely, 1in., $1\frac{1}{4}$in., $1\frac{5}{8}$in. and $2\frac{5}{8}$in. capacity. A $1\frac{1}{4}$in. machine is shown in Fig. 6. The American construction of these machines, whereby the frame comprises only four castings, pan, headstock, gearbox and top rail, gives a streamlined appearance and at the same time encloses all mechanism to give protection against the infiltration of swarf and dirt. They are essentially heavy duty machines, and are designed to withstand heavy cuts without vibration. In order that optimum cutting speeds may be used on different applications and materials each size has a wide range of spindle speeds, the actual figures being :—

1in. capacity : 719 to 2383 r.p.m.
$1\frac{1}{4}$in. capacity : 252 to 2181 r.p.m.
$1\frac{5}{8}$in. capacity : 108 to 1380 r.p.m.
$2\frac{5}{8}$in. capacity : 101 to 1065 r.p.m.

The tool slide is made from heat-treated high carbon steel, and the surfaces which are used for tool holders are ground to ensure smooth action. Renewable bronze bushings and a felt wiper are fitted to the main tool slide, which also has a positive stop to control such functions as the depth of drilling or reaming and length of turning. Combination tooling is easily arranged. The tool slide guide is mounted on a bracket underneath the slide so that the 3rd, 4th and 5th positions are left free for the independent operation of threading and accelerated drilling and reaming attachments. To ensure alignment of tool holders and spindle, the holders are mounted on the slide before they are bored out from the spindle. Permanent correlation is ensured by mounting the slide round a stem that is integral with the spindle carrier and then grinding both stem and carrier at one setting.

The two main cross slides are usually wide and long. They move on hardened steel ways mounted on the headstock. This firm mounting allows heavy forming cuts to be taken at high speeds without causing vibration or chatter. Powerful direct action is provided from a cam drum beneath each slide and by an independent stop in each slide which engages an adjustable stop on the periphery of the spindle carrier. When small adjustments for depth of cut are required it is not necessary to interfere with the tool holders, since an adjusting screw is provided for this purpose. It can be locked after being set. The design of the cross slides in the first and sixth position is such as to prevent the entrance of chips into the cross slide bearings or ways. A similar arrangement of stops is used for the two top cross slides.

Drive to the spindle is by vee ropes from a motor mounted on an adjustable platform above the headstock. For convenience in setting up there are push buttons for "start", "stop" and "inch" at both the front and the back of the machine. The change from approach to cutting traverse is effected by cone and plate type clutches actuating a cam. This, in conjunction with a brake, allows very fine adjustments to be made. Speed and feed changes are made through pick-off gears at the main drive end of the machine. Since these gears are interchangeable for both speeds and feeds, only a small number is required. A modified Geneva mechanism is used to index the spindle carrier.

Chucking and feeding slides are mounted on two large diameter steel shafts supported in the headstock at one end and in an arch support at the other. These slides have long bearing surfaces on the shafts to obviate any tendency to tip or bind. The chucking shoes incorporate wide bearing surfaces on the chucking spools. Provision is made to eliminate any possibility that chuck release will occur during machining. The cams for operating the chucking and feeding mechanism are conveniently arranged on a drum located directly under the slides. This facilitates quick change and removal of end stock.

RESEARCH IN GERMANY

THE Department of Scientific and Industrial Research, Charles House, 5-11, Regent Street, London, S.W.1, have issued translations of a series of reports on research work carried out under contract in Germany under the heading "Sponsored Research (Germany)". Report No. 1 gives the results of a practical investigation of the relation between surface roughness and the alternating bending strength of steel, and indicates the probable existence of a simple relationship between the two. Reports No. 3, 4, 5 and 6 contain the results of a theoretical investigation of the deformation of loaded gears, the ultimate object being improved design of gears. The report No. 11 gives details of the processes used to compute the pressure distribution in a 360 deg. journal bearing, and Report No. 13 gives an outline of the process proposed for the computation of the pressure distribution in a similar bearing with an inclined shaft. A further report, which will be of interest to production engineers as well as test laboratory and research staff, is Report No. 2. This is a description of equipment used to investigate the rolling strength of different steels. The prime objective of this research was to find the causes of the phenomenon known as "pitting". It may be noted that with the exception of Report No. 1 and 2 the work is of a fairly involved mathematical nature, but all the reports show evidence of an extremely high standard of work and could well be enlarged upon by the scientific departments in industry, and interpreted for practical application to design.

(1945)

Miniature Ball Bearings

ALTHOUGH miniature ball bearings are not employed on the motor vehicle as such, they are finding more and more application in instrument work. In the design of testing and recording apparatus, the miniature ball bearing has now made for itself an essential niche, as it has of course in many other applications, as for example, clocks, dental apparatus, metering devices, etc. These bearings are in fact obtainable in such minute dimensions as 1.1 mm. diameter for bearings of the pivot type, and they are now marketed in all sizes and forms by Miniature Bearings, Ltd., 192, Sloane Street, London, S.W.1., who are the sole British importers of the famous Swiss R.M.B. miniature bearings.

(1942)

Rubber in Engineering

THE Andre Rubber Co. Ltd., Kingston By-pass, Surbiton, Surrey, have recently issued a brochure showing some of the many ways in which rubber can be used in engineering products. The first section deals with metal to rubber bonded parts, such as resilient gears, brake rod guides, engine mountings, rubber seals, and including a coupling to transmit 1,100 h.p., and a sandwich type rubber spring to support 60 tons. The second part illustrates moulded rubber parts such as control knobs, universal joint covers, trolley castors, and tail light bodies, whilst the last section shows components covered with a protecting layer of rubber for abrasion resistance, or resistance against certain corrosive fumes and liquids, etc. The publication is of particular interest to engineers seeking new ideas on the ever expanding field of application of rubber to industrial products. Sixty-six full page and three-quarter page illustrations are included.

(1943)

The Institution of Mechanical Engineers, Automobile Division

AUTOMOBILE PETROL ENGINES

A Survey of Factors Influencing Performance and Weight

By Donald Bastow, B.Sc. (Eng.), M.I.Mech.E.*

This paper gives the results of a general survey of the influence of various factors, such as valve arrangement, head design, octane number, and specific weight, on the power output of automobile engines. The scope of the paper is similar to that of the very early one presented to the Institution of Automobile Engineers (Lanchester 1907)†, but has benefited from the results of recent work.

It was felt that many of the conclusions drawn from the survey could be presented most effectively by calculating, as a basis for comparison, the primary dimensions and performance characteristics of a series of engines, each intended to develop 100 b.h.p. The results are collated in tabular form.

IN a general survey of this kind there is often, and there was in this instance, a choice between a necessarily limited range of information of which the accuracy is beyond question because it is the product of personal experience, and a much wider range whose accuracy is suspect because it has been published *via* a sales organization. To broaden the scope of the investigation it was considered essential to use a large amount of information from this source (Anon. 1949). In making predictions for personal use, some allowance must be made for the source of the information. Apart from the question of exaggeration, however, differences in engine-output figures may arise from differing methods of taking observations.

Component weights are always a matter of concern to the automobile engineer, and the present survey sprang from an attempt to classify a representative collection of American passenger-car engine and gearbox weights (Riesing 1950). It was broadened with a limited number of British counterparts. Although first chronologically, the weight aspect fits most conveniently at the end of the paper, and is therefore placed there.

Reasons for Variations in Engine Performance

The performance of an engine is most completely expressed by the brake mean effective pressure (b.m.e.p.) curve. This is the indicated mean effective pressure (i.m.e.p.) curve less the curve of frictional losses. The i.m.e.p. varies with the cylinder size, compression ratio, volumetric efficiency, type of combustion chamber, valve arrangement and fuel. Frictional losses arise from bearing and piston friction, and pumping losses. Poor aspiration, or "breathing", therefore results in a double loss; an increase in friction because more work is required to introduce the gases into the cylinder (and possibly to reject them afterwards), and a reduction in the quantity of gases dealt with per cycle and hence of the output. This is almost certainly the more important aspect of the two.

Apart from the b.m.e.p., the maximum power developed by an engine depends on the maximum piston-speed attained. There has been a tendency to regard this as being independent of engine proportions and size, but evidence will be produced suggesting that this is not true.

Friction Loss Variation. It is desirable to consider friction losses first, because, by obtaining a reasonable approximation to them, results derived from b.m.e.p. figures, which comprise the bulk of the available information, may be compared with others which are given as i.m.e.p. figures. For this purpose accurate figures for the friction m.e.p. are unnecessary, as large percentage variations in them make only small percentage differences to the b.m.e.p. and i.m.e.p. figures derived by their use. This is fortunate because of the limited information which is available. From the work of Roensch (1949) it is known that the friction m.e.p. varies with the compression ratio; piston friction and pumping losses would be expected to vary with piston speed (other factors being equal), and bearing losses would be expected to vary with the crankshaft speed. Accordingly, those curves which are available have been plotted against both crankshaft speed and piston speed (Fig. 1). The curves appear to group better when plotted against crankshaft speed—the curves for the 3-litre T.T. Vauxhall engine (Ricardo 1922) are an example. However, the 3-litre Vauxhall engine would be expected to have lower friction m.e.p. figures than other engines, because it has roller main- and big-end bearings. In general it seems fair to infer that engines with small crankshafts and free breathing have friction m.e.p. figures lower than the average. No explanation can be given for the high friction values of the 1912 Adler engine (Riedler 1914).

Suggested curves of friction m.e.p. against both piston speed and crankshaft speed have been derived from the plotted curves of Fig. 1 and are reproduced in Fig. 2. Most weight has been given to the values quoted by Roensch (1949) and to those from the experimental engine. In estimating the friction m.e.p. for any engine, the method is to take the arithmetic mean of the values derived from these curves as the true figure for a particular

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†An alphabetical list of references is given in the Appendix.

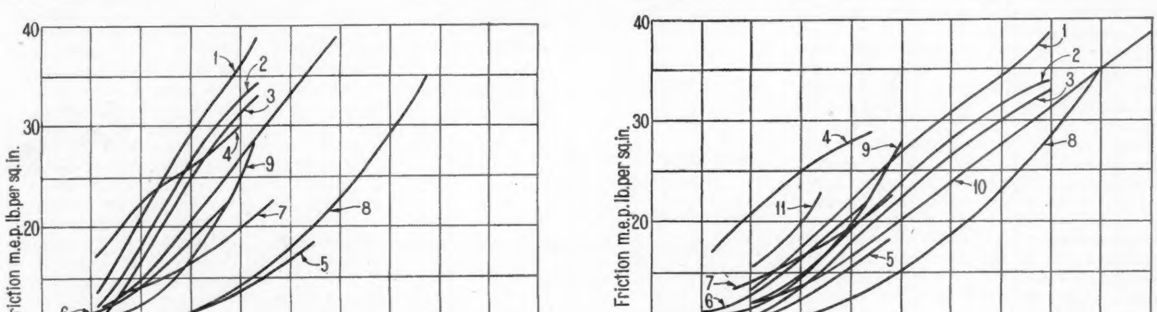

Fig. 1. Variation of friction m.e.p. with (a) piston speed, (b) crankshaft speed.

Curves 1—3 are derived from Roensch; 4—7 from Riedler; 8 and 9 from Ricardo. Curve No. 10 relates to an experimental engine, and curve No. 11 (Fig. 1*b*) is derived from Tookey.

Curves: 1. Compression ratio 12/1. 2. Compression ratio 10/1. 3. Compression ratio 8/1. 8. Compression ratio 5.8/1. 9. Compression ratio 5.1/1. 10. Compression ratio 6.5/1.

Table I. Variation of I.m.e.p. with the Index of *d*

Condition	I.m.e.p. at 2.25 inches bore, lb. per sq. in.	I.m.e.p. at 4.0 inches bore, lb. per sq. in.
I.m.e.p. $\propto d^{0 \cdot 10}$	145.75	154.38
I.m.e.p. $\propto d^{0 \cdot 15}$	143.67	156.61
I.m.e.p. $\propto d^{0 \cdot 20}$	141.61	158.88

engine. The curves take account of the variation in friction m.e.p. with compression ratio.

Roensch's figures suggest that the friction m.e.p. varies with compression ratio in the same proportion as the b.m.e.p. or, expressed in another way, that the mechanical efficiency of an engine does not alter with changes in compression ratio. This is important, because it means that both the i.m.e.p. and the b.m.e.p. vary in the same way with, for instance, changes in cylinder size or compression ratio.

Variation of I.m.e.p. with Cylinder Bore. For some time it has been accepted that there is variation of i.m.e.p. with cylinder size, defined either by the bore or swept volume. It is immaterial which is used providing the stroke/bore ratio is constant for the engines considered. Unfortunately this ratio varies fairly widely.

David and Leah (1940) consider the cylinder bore, rather than the swept volume, as the basis from which to calculate heat losses during combustion and expansion. The variation of the i.m.e.p. with the cylinder bore at varying crankshaft speeds, prepared from information presented by David and Leah, is shown in Fig. 3. Octane fuel, 120 per cent. mixture strength, and 75 per cent. volumetric efficiency, are assumed. The curves relate to a compact combustion chamber. A combustion chamber of less compact form, for example, a "turbulent" chamber, would have higher heat-losses and therefore a greater variation of i.m.e.p. with cylinder bore.

The b.m.e.p's. for several groups of engines, each group having the same compression ratio and generally similar characteristics, were plotted against cylinder bore and swept volume. The b.m.e.p.-bore curves were more consistent, and this relationship has accordingly been taken as a basis for calculation.

Logarithmic plotting shows that, over the range from 2.25 to 4.0 inches bore, the results correspond reasonably well with the relationship : i.m.e.p. $\propto d^{0 \cdot 2}$, where d is the cylinder bore. This rate of change is about twice that derived from the work of David and Leah (1949) for a compact combustion chamber and a speed of 1,000 r.p.m. Most of the combustion chambers of the engines which gave the approximate relationship i.m.e.p. $\propto d^{0 \cdot 2}$ were far from compact. It is possible that there are sources of variation of i.m.e.p. with cylinder bore, other than the variation due to changes in heat losses, for example, there could be a possible increase in volumetric efficiency with increased bore, or a reduction in friction m.e.p.

Results given by Ricardo (1922) for a group of three engines of increasing size, and with compact combustion chambers, give an average value of 0.15 for the index of d.

In view of the uncertainty of the slope of a line derived from a number of scattered points it is safer to take a value for the index which is a good compromise of the various indications ; on this basis the relationship i.m.e.p. $\propto d^{0 \cdot 15}$ is assumed.

Fairly large variations in the index of d, with which the i.m.e.p. is assumed to vary, do not result in impossibly large variations of the i.m.e.p. values. When the index is as much, say, as 0.05 in error, and extreme cases are considered, by going from an assumed i.m.e.p. of 150 lb. per sq. in. at 3-inch bore to the lower and upper limits of 2.25- and 4.0-inch bores, the results are as shown in Table 1.

Variation of I.m.e.p. with Compression Ratio. The thermal efficiency figures and formulae given by Leah (1949) enabled curves of i.m.e.p. against compression ratio to be derived for the three fuels, octane, benzene, and ethyl alcohol. These are shown in Fig. 4*a* for the assumed conditions of 75 per cent. volumetric efficiency and 100 per cent. mixture strength. The octane curve is repeated at volumetric efficiencies of 70 and 65 per cent. for reference purposes. The figure from which the curves are derived are based on the assumption that no heat is lost, and, because they take into account dissociation, they appear as curves even with logarithmic plotting.

Ricardo (1922) carried out some comprehensive tests with different fuels, using a variable compression-ratio engine, and by

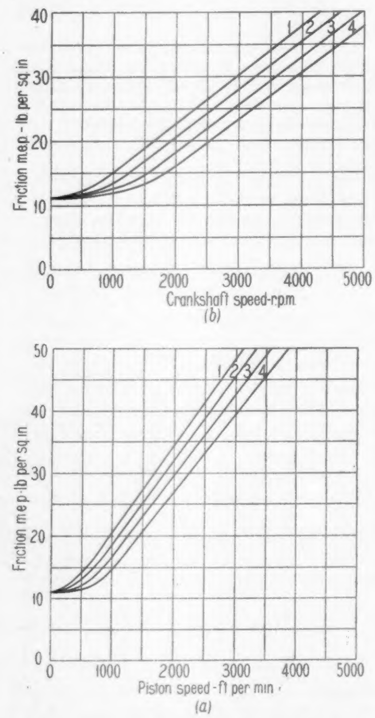

Fig. 2. Suggested curves of variation of friction m.e.p. with (*a*) piston speed, (*b*) crankshaft speed.

Curves : 1. Compression ratio 12/1.
2. Compression ratio 10/1.
3. Compression ratio 8/1.
4. Compression ratio 6/1.

extracting from them only those tests in which the results were not seriously affected by the fuel used, a series of values of i.m.e.p. at different compression-ratios were obtained (Fig. 4*a*). These curves show less variation with compression ratio than Leah's results would lead one to expect. Ricardo (1922) mentions that it had been observed that the volumetric efficiency fell off as the compression ratio was increased, thus accounting for the disparity.

In other instances, however, where different compression ratios are used in the same engine, there does not seem to be the same reduction of volumetric efficiency with increase in compression ratio. The two points for Buick (Anon. 1949) engines of 6.3/1 and 6.6/1 compression ratio both lie on the 69 per cent. volumetric efficiency octane curve, the points for the high-compression engine results given by Roensch (1949) also seem to follow the slope of the octane curve reasonably well. There appears to be, therefore, reasonable justification for assuming that this curve should be taken as the basis of variation with compression ratio. Over the range of compression ratios from 6.0/1 to 8.0/1, the variation can be expressed as i.m.e.p. $\propto r^{0 \cdot 365}$, where r is the compression ratio, without undue error. This is for octane ; corresponding indices for benzene and ethyl alcohol are 0.41 and 0.391 respectively. It is believed that the octane curve more nearly represents what is obtained with normal petrol.

It is interesting to consider whether variation in volumetric efficiency with compression ratio is to be expected. Volumetric efficiency depends on the quantity and temperature of the residual gases, the temperature of the incoming mixture, and the pressure in the cylinder at the end of the suction stroke. The quantity of residual gases depends on the pressure at the end of the exhaust stroke and, the clearance volume (or compression ratio). Assuming an inlet-gas temperature of 60 deg. C. (140 deg. F.) ; a residual-gas temperature of 600 deg. C. (1,400 deg. F.) ; a compression ratio of 6/1 ; equal specific heats for the residual gases and inlet gas, and atmospheric pressure at the end of the exhaust and suction strokes, a volumetric efficiency of 81.7 per cent. is obtained, with a temperature at the end of the suction stroke of 99 deg. C. (210 deg. F.). A variation of 100 deg. C. (180 deg. F.) in the temperature of the residual gases affects the answer by as little as 0.3 per cent., and the temperature at the end of the suction stroke by as little as 4 deg. C. (9 deg. F.). Increasing the compression ratio of 8/1, leaving the other assumptions unaltered, gives a volumetric efficiency of 82.2 per cent. and a suction temperature of 87 deg. C. (189 deg. F.). A drop of 100 deg. C. (180 deg. F.) in the residual gas temperature would then reduce the volumetric efficiency to 82 per cent. and the suction temperature to 85 deg. C. (185 deg. F.). There does not seem, therefore, to be any fundamental reason for considerable variation in volumetric efficiency with compression ratio.

The next stage is to take the i.m.e.p. figures, obtained from the given maximum b.m.e.p. values for various engines by adding to them the friction m.e.p. figures from the "suggested" curves of Fig. 2 and to plot them logarithmically against compression ratios. The results are given in Fig. 4*b*. As guides there are lines of 75, 70, 65, and 60 per cent. volumetric efficiency, for octane fuel, as in Fig. 4*a*, taken from Leah's results. To eliminate size effect all the i.m.e.p's. are reduced to those for a "reference" bore of 3.0 inches

on the assumption that i.m.e.p. $\propto d^{0\cdot15}$. The figures for hemispherical head engines lie, in general, within the range 72-76 per cent. volumetric efficiency, although there is one as low as about 68 per cent. The average lies at about 74 per cent. In-line overhead-valve engines (with which, perhaps illogically, are included hybrids like the F-head, or overhead inlet, side exhaust-valve arrangement and the lozenge head) have a much wider spread, between about the 61 and 73 per cent. volumetric efficiency lines, with an average of about 68 per cent. The American engines, often referred to as "woolly", are rather above this average. Side-valve engines also have a wide spread, from the 60 to the 68 per cent. lines, with one or two strays below this range. The average seems to lie at about 63 per cent.

Although the various points are referred to in terms of their position in relation to lines of a given volumetric efficiency derived from Leah's thermal efficiencies, these are not the actual volumetric efficiencies obtained in the engines in question. The i.m.e.p. is adversely affected, as compared with the ideal values from Leah's (1949) thermal efficiencies, by reduced volumetric efficiency and by heat losses to the combustion space, and may be affected either way by changes in mixture strength. In the absence of either volumetric-efficiency curves or fuel-consumption figures it is not possible to determine the relative importance of the different factors. Ricardo (1922) gives both i.m.e.p. and volumetric-efficiency figures for the 3-litre T.T. Vauxhall engine. For the compression ratio of 5.8/1, and the stated volumetric efficiency of 80.7 per cent., is derived, from Leah's figures, an i.m.e.p. of between 162 and 163 lb. per sq. in. for both octane and benzene, and the maximum observed i.m.e.p. is 162.5. This implies that there are no heat losses, which is impossible, and the explanation of this inconsistency is not immediately obvious. Leah (1949) states that his results are true for a suction temperature

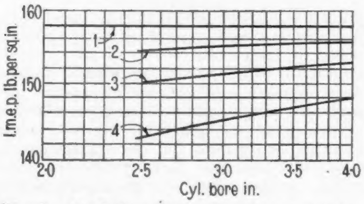

Fig. 3. Variation of i.m.e.p. with cylinder bore.

Curves: 1. Ideal value.
2. Attainable—5,000 r.p.m.
3. Attainable—2,000 r.p.m.
4. Attainable—1,000 r.p.m.

of 100 deg. C. (212 deg. F.) and that a correction must be made for temperatures other than this. Curves for the evaluation of this correction are given by David and Leah (1940), and according to these a drop in temperature of 37 deg. C. (67 deg. F.) at the end of the suction stroke would be required to obtain an increase of 1 per cent. in the i.m.e.p. It seems unlikely, therefore, that this can be the only, or even the main, source of the discrepancy. The opinion has been formed, however, that the curves of performance of the Vauxhall engine have been "smoothed".

David and Leah stated that it was generally agreed that indicated thermal efficiencies and i.m.e.p's. as determined by the motoring method, were overestimated, because the piston friction and the pumping losses measured during the motoring test overestimated those in the engine when running under power. This seems to be the most probable main reason for the apparent absence of heat losses suggested by the results quoted, resulting in the conclusion that the i.m.e.p. values quoted for the 3-litre T.T. Vauxhall engine are higher than those obtained. This fact has been ignored in deriving the average friction m.e.p. values of Fig. 2.

Maximum Piston Speed. Lanchester (1907) demonstrates by dimensional theory that the horse-power of engines of different sizes, not necessarily geometrically similar, varies as the square of the linear dimensions. The full equation is h.p. $= \sigma\phi^{0\cdot5}l^2 \times$ a constant.

Here, σ represents the "stress" in the working fluid, that is, the mean effective pressure, and ϕ is the specific strength, or the working stress divided by the density, of the material of construction.

He shows further that if the bore and stroke are d and s respectively, h.p. $= \sigma\phi^{0\cdot5}d^n s^{2-n} \times$ a constant. From the relationships $\sigma \propto r^{0\cdot365}d^{0\cdot15}$ and h.p. $\propto \sigma \times$ piston area $\times$ piston speed, the expression h.p. $\propto \sigma \times d^2 \times$ piston speed is obtained, as the piston area varies as d^2.

There is a popular belief that the maximum piston-speed is constant regardless of engine size and proportions, but there are reasons for disbelieving this. The first indication of this was obtained when an investigation was made, starting with an engine of 84 mm. bore and 90 mm. stroke, into the effects of shortening and lengthening the stroke. Dimensions of 67.5 and 115 mm. were taken as the two extremes, and the dimensions of the connecting rods and cranks were altered to maintain a constant ratio of connecting-rod length of stroke and to preserve the required crankshaft rigidity. Two criteria were used to determine the maximum piston-speed in each case, the speed at which the PV factor of the big-end bearing was the same as in the original engine, and the speed which bore the same proportion to the natural torsional-periodicity of the crankshaft assembly. No difficulty was found in making these identical, and it was discovered that the longer stroke permitted a higher piston-speed and the shorter stroke required a lower piston-speed.

In the discussion on a paper on crankshaft torsional vibrations by Ker Wilson (1936), Llewellyn Smith stated that, in connexion with the theoretical frequencies of the crankshaft of a certain $3\frac{1}{2}$-litre engine, it had been found that if the stroke were varied and the cylinder centre distances and

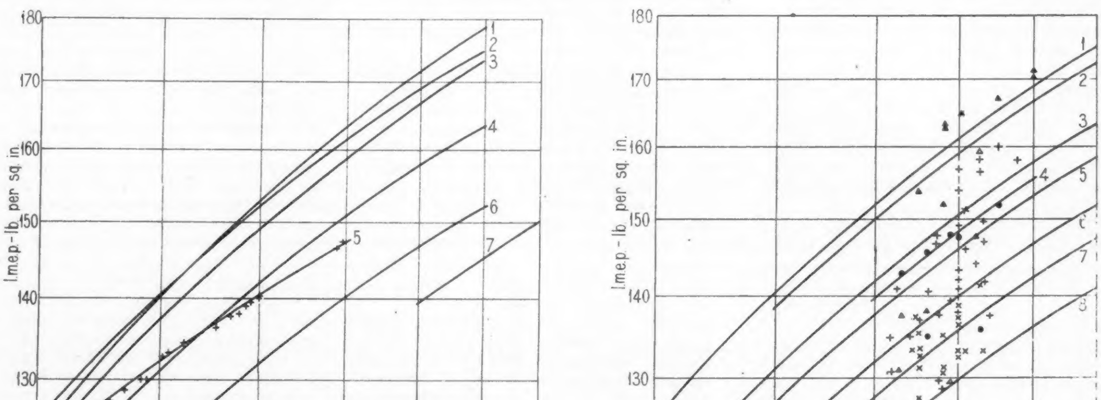

Fig. 4. Variation of i.m.e.p. with compression ratio.

a Derived from Leah and Ricardo.
Curves: 1. Benzene } 75 per cent. volumetric efficiency.
2. Octane
3. Ethyl alcohol
4. Octane—70 per cent. volumetric efficiency.
5. Ricardo.
6. Octane—65 per cent. volumetric efficiency.
7. Roensch.

b. Derived from the curves of Fig. 2.
Curves: 1. Octane—75 per cent. volumetric efficiency.
2. Suggested average—hemispherical head.
3. Octane—70 per cent. volumetric efficiency.
4. Octane—69 per cent. volumetric efficiency.
5. Suggested average—in-line overhead-valve engines.
6. Octane—65 per cent. volumetric efficiency.
7. Suggested average— side-valve engines.
8. Octane—60 per cent. efficiency.
▲ Hemispherical-head engines.
● In-line overhead-valve engines (American).
× Side-valve engines (American).
+ In-line overhead-valve engines (British).
⊿ Side-valve engines (British).

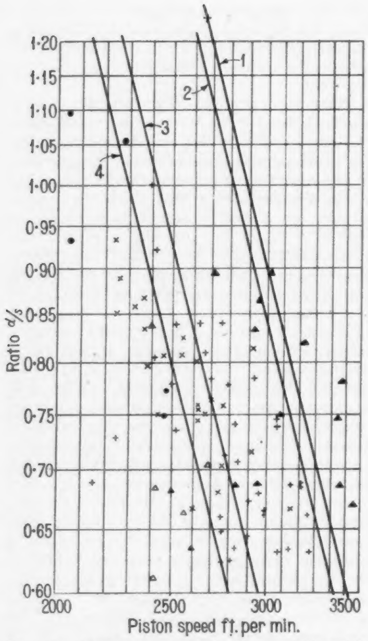

Fig. 5. Variation of piston speed with engine proportions.

Curves : 1. Line through "datum" points.
2. Suggested line—hemispherical heads.
3. Suggested line—in-line overhead-valve engines.
4. Suggested line—side-valve engines.

▲ Hemispherical-head engines.
● In-line overhead-valve engines (American).
× Side-valve engines (American).
+ In-line overhead-valve engines (British).
Δ Side-valve engines (British).

bore were kept constant, the critical periods of the engine would always occur at constant piston-speeds. If this conclusion were generally applicable, it would be important, because it would mean that the position of the critical speeds would always be the same in relation to the power curve of the engine, whether the stroke were increased or decreased, provided that the bore and length of the engine were unaltered. This disagrees with the results previously mentioned, but the assumptions made by Llewellyn Smith in arriving at the results which led to his statement are not known.

From the form of the expression already obtained, the piston speed must vary, either with a non-dimensional property, or with d/s. Accordingly the values for the piston speed at maximum power, of the various engines of which particulars were available, were plotted logarithmically against the ratio d/s (Fig. 5), the three points corresponding to hypothetical engines being inserted as a guide, and the engines again being divided into three categories, side valve, in-line overhead valve, and hemispherical head. Average lines for the three sets of points which were parallel to the line through the "datum" points could be drawn, and for a ratio of d/s of unity, corresponded to piston speeds of 2,250, 2,430, and 2,810 ft. per min. respectively. The slope of the lines gives the relationship piston speed $\propto (d/s)^{0.35}$. If this is put into the expression for the power, b.h.p. $= \sigma\phi^{0.5}d^{1.65}s^{0.35} \times$ a constant.

Maximum Brake Horse-power. Including in the foregoing expression the value already obtained for the mean effective pressure, σ, and for general convenience assuming that the materials of construction do not vary (so that ϕ may be included in the constant term), the equation

$$\text{b.h.p.} = \text{constant} \times r^{0.365}d^{1.8}s^{0.35}$$

is obtained. For average values, different values of the constant must be used for side valve, in-line overhead valve, the hemispherical-head engines, and for more particular evaluation, differences in material for connecting rods, pistons, bearings, and so on, and differences in breathing capacity, must be allowed for.

Variation of Compression Ratio with Bore. Picard (1949) gives a graph showing the variation of permissible compression-ratio with bore for fuels of different octane numbers. He states that the curves have been derived from results given by a number of known engines ; no information is given as to the types of cylinder head, nor is any suggestion made that the results might be affected by different types of cylinder head and valve arrangements. If the results are accepted as being applicable to any type of head, and applied to graphs of i.m.e.p. against cylinder bore for different types of engine and different compression ratios, two conclusions may be drawn. They are :—

(1) At low octane numbers (65 and 70) the maximum realizable i.m.e.p. for a fuel of given octane number, and a given type of combustion chamber, is almost entirely independent of the bore. To achieve this result it is necessary to increase the compression ratio as the bore is decreased. At higher octane numbers (75 and 80) the attainable i.m.e.p. appears to be less for larger cylinder bores, the reduction in

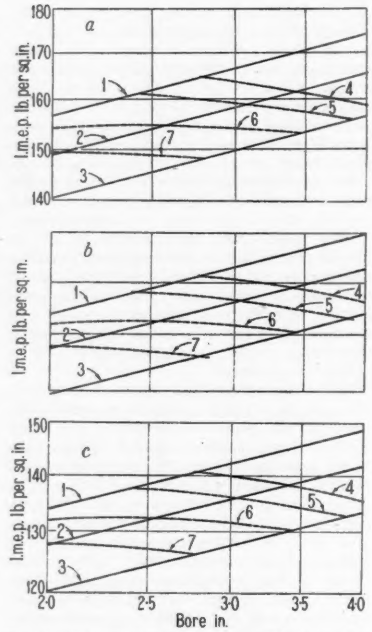

Fig. 6. Variation of i.m.e.p. with bore, for different compression ratios and octane numbers.

a Hemispherical heads.
b In-line overhead valves.
c Side valves.

Curves : 1. Compression ratio 8/1.
2. Compression ratio 7/1.
3. Compression ratio 6/1.
4. 80 octane.
5. 75 octane.
6. 70 octane.
7. 65 octane.

compression ratio necessitated by the larger bore having predominated over the increase of i.m.e.p. with bore at a given compression ratio.

(2) The highest mean effective pressure for a fuel of given octane number is obtained by using a hemispherical head, and the lowest from a side-valve arrangement, with the in-line overhead valve intermediately.

With regard to the first conclusion, the picture as far as b.m.e.p. is concerned may be somewhat different because of variation in friction m.e.p. with cylinder size. There appears to be some evidence that friction m.e.p. is less for engines of larger bore (for instance, the results for the range of three

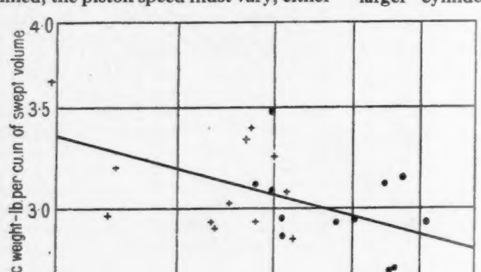

Fig. 7. Specific weight of 1949 American passenger-car engines plotted against swept volume.

+ Six-cylinder engines. ● Eight-cylinder engines.
The unit weights include engine and gearbox.
The average line is drawn in.

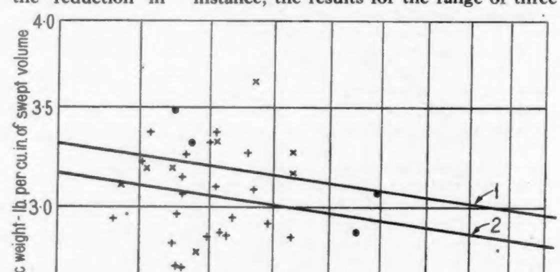

Fig. 8. Specific weight of 1949 American passenger-car engines plotted against bore/stroke ratio.

Curves : 1. Average line—British engines.
2. Average line—American engines.
+ American engines. ● American overhead-valve engines. × British engines.
The unit weights include engine and gearbox.

Table II. Particulars of Specimen Engines

	Engines for 70-octane fuel				Engines for 80-octane fuel			
	Bore/stroke ratio, 0.70/1			Stroke, 2.66 inches	Bore/stroke ratio, 0.70/1			Stroke, 2.66 inches
	Side valves	In-line overhead valves	Hemi-spherical head	Hemi-spherical head	Side valves	In-line overhead valves	Hemi-spherical head	Hemi-spherical head
Number of cylinders ..	6	6	6	6	6	6	6	6
Bore	3.4 inches (86.3 mm.)	3.2 inches (81.3 mm.)	2.875 inches (73 mm.)	3.15 inches (80 mm.)	3.312 inches (84 mm.)	3.11 inches (79 mm.)	2.775 inches (70.5 mm.)	3.03 inches (77 mm.)
Stroke	4.85 inches (123 mm.)	4.56 inches (116 mm.)	4.1 inches (104 mm.)	2.66 inches (67.5 mm.)	4.725 inches (120 mm.)	4.44 inches (112.8 mm.)	3.96 inches (100.5 mm.)	2.66 inches (67.5 mm.)
Capacity, cu. in. .. .. litres	265 4.34	220.6 3.62	160 2.62	124.4 2.04	244 4	202.4 3.32	144 2.36	115 1.885
Compression ratio ..	6.0/1	6.31/1	6.66/1	6.37/1	7.1/1	7.45/1	8.0/1	7.57/1
Maximum b.h.p. ..	100	100	100	100	100	100	100	100
Speed at maximum b.h.p., r.p.m.	3,250	3,650	4,700	5,900	3,350	3,750	4,950	6,050
Piston speed at maximum b.h.p., ft. per min. ..	2,620	2,770	3,220	2,615	2,640	2,770	3,220	2,670
Maximum b.m.e.p., lb. per sq. in.	114.5	123	132.5	132.5	122.0	131.0	141.5	140.4
Maximum torque, lb.-ft. ..	202	181	141	110	198	177	135.5	107.5
Weight of engine-gearbox, lb.	855	734	569	428	794	681	515	404
Weight saving from side-valve unit, lb. ..	—	121	286	427	61	174	340	451
Attainable fuel consumption, lb. per b.h.p. hr. ..	0.513	0.50	0.491	0.499	0.48	0.471	0.462	0.470
Relative miles-per-gallon index	1.0	1.059	1.127	1.158	1.085	1.140	1.217	1.237
Fuel consumption, m.p.g.	20.0	21.78	22.54	23.16	21.7	22.4	24.34	24.74

engines given by Ricardo (1922) to which reference has been made earlier) and if this is generally true, the higher b.m.e.p. will be obtained from a larger-bore engine, for a fuel of low octane number and a given type of combustion chamber. For higher octane-number fuels (80 for instance) it seems unlikely that variations in friction m.e.p. with size can outweigh the drop in i.m.e.p. with increase in bore shown in Fig. 6.

Until there is more experimental confirmation of the assumption that the maximum compression-ratio for a fuel of given octane number and for a given cylinder bore is independent of the type of combustion chamber, it would be unwise to consider the second conclusion as being definite. If the assumption is correct, there are advantages in using the hemispherical cylinder head when the octane number of the available fuel is limited.

If, in addition, the assumptions on friction m.e.p. are also true, the highest mechanical efficiency should be realized from an engine with a hemispherical cylinder head and, other factors being equal, an engine with this type of cylinder head should also give the best specific fuel-consumption.

More information on the way in which allowable compression-ratio varies with the cylinder bore, octane number, and type of cylinder head, would be useful.

Weights of Engine-Gearbox Units

Riesing (1950) gives the weights of the 1949 American passenger-car engine and gearbox units. The best way to express the weights of these units is as specific weights, in units of lb. per cu. in. of swept volume. It was anticipated that there would be some size effect because of the effect of accessory weights and the limiting casting-thickness. To investigate this the specific weights were plotted logarithmically against the engine displacement (Fig. 7). There is considerable scatter of the points, but the tendency to increase in specific weight with decrease in displacement is clear. The average line through the points gives the relationship specific weight (lb. per cu. in. of swept vol.) $\propto$ displacement$^{-0.1775}$.

It was expected that there would be some variation of specific weight with the bore/stroke ratio, and the specific weights reduced by the above relationship to those for a "reference" displacement of 250 cu. in. were next plotted logarithmically against the ratio d/s (Fig. 8). Here also, in spite of the scatter, there is seen to be a tendency for the specific weight to decrease as the ratio d/s increases, and the slope of the mean line suggests the relationship, specific weight $\propto (d/s)^{-0.1235}$.

For the American engine-gearbox units of which Riesing gives particulars, the combination of these two curves gives the relationship:—

Specific weight $= 2.97 \times (d/s)^{-0.1235} \times$ displacement/250)$^{-0.1775}$.

The overhead-valve engines, with the exception of the Cadillac and Chevrolet, lie above the average line.

A limited number of weights of British engine- and gearbox-units were available, and these also give an average lying above the American average. For the British engines of which particulars are available

Specific weight $= 3.1 \times (d/s)^{-0.1235} \times$ (displacement/250)$^{-0.1775}$.

All of the engines included in this investigation had cast-iron cylinder blocks and heads; differences would be expected if either of these main components were made of other materials.

The specific weight depends, to some extent, upon the ratio of connecting-rod length of stroke, and any variation in this from an average figure must also affect the specific weight of the engine.

Petrol Consumption

From, for example, Fig. 3, if the speed, cylinder bore, compression ratio, fuel, volumetric efficiency, and mixture strength are known, the indicated thermal efficiencies and i.m.e.p. values may be obtained from information given by David and Leah (1940). From these, and the assumed friction m.e.p. values given in Fig. 2, it is a simple step to arrive at attainable brake consumptions for any given engine.

Conclusions

Many of the conclusions to be drawn from this survey appear to be expressed most clearly by calculating the major particulars of a series of engines, each intended to develop 100 b.h.p. This series is divided into two groups, the first to run on 70-octane fuel, the second on 80 octane. The first three engines in each group have a constant bore/stroke ratio of 0.70/1, and have respectively, side valves, in-line overhead valves, and overhead valves in a hemispherical head, while the fourth in each group also has overhead valves in a hemispherical head, but the stroke has been reduced to what is considered to be the shortest practicable for a connecting-rod length/stroke ratio of 1.8/1. The main particulars of these engines are given in Table 2.

The attainable specific consumptions are based on octane fuel, the maximum-torque speed (taken as half the full-power speed in each case), and 120 per cent. correct mixture-strength, the heat losses given by David and Leah (1940) for a compact combustion chamber being assumed to apply in each case, although clearly there should be variations between the different types of combustion chambers. The values of the fuel consumption (miles per gallon) (m.p.g.) index are based on these (full throttle) specific-consumption figures. Further assumptions of constant ton-m.p.g., when fitted with the side-valve engine to run on 70-octane fuel, and a vehicle weight, including driver and passenger, of 3,800 lb. are made. The vehicles with the alternative engines are assumed to be reduced in weight only by the amount of the weight saving on the engine-gearbox unit. In practice a reduced engine-weight should allow further consequential weight savings and so imply even more improvement in fuel consumption (m.p.g.) with reduction in engine weight than Table 2 indicates. In order to make the relative m.p.g. values more easily understandable they have been shown also as actual values, based on an assumed value of 20 m.p.g. with the side-valve unit running on 70-octane fuel.

Acknowledgments. Acknowledgments are due to Mr. W. O. Bentley, since it was during the Author's employment by him that the paper was written; and to the B.S.A. Group Research Organisation for permission to publish the paper.

APPENDIX

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CYLINDER BORES

A New Treatment to Cover the Running-in Process

During the running-in and testing of any newly assembled automobile engine, it is usual to run for several hours under no load or light load conditions. On completion of test, an engine will usually stand for a short period, and it is found that on removal of the cylinder head, the cylinder bores bear evidence of rusting. This is thought to be due to condensation of water and to the presence of small quantities of corrosive compounds from the fuel and elsewhere that are dissolved in the water.

The presence of even the smallest traces of such rust is obviously detrimental to the ultimate life of the engine, and yet it should be noted that, in the case of most private and commercial units, these engines are assembled in chassis and put to work whilst this rusting is still present. The initial abrasive action can result in permanent scoring of pistons and cylinder walls.

That this state of affairs does not exist throughout the life of the engine shows that the correct running-in of the cylinder walls eventually eliminates this trouble, but not before the damage and undue wear has taken place. Running-in will eventually provide a more perfect oil-retaining surface which will prevent such corrosion, but up to now it has been difficult, if not impossible, to secure this oil-retaining surface during the most carefully controlled honing of cylinder walls.

A considerable amount of work over a long period has been undertaken by British Vapour Blast Ltd., Central Chambers, Market Square, Wellington, Shropshire, under the sponsorship of one of the leading British motor manufacturers, with the object of overcoming this initial cylinder wall corrosion. Several blocks have been treated by a special form of vapour blasting and in all cases rusting has been entirely eliminated.

The surface generated by vapour blasting is a fine capillary one. Early investigations showed that a tiny spot of lubricating oil dropped on a piece of metal which had been vapour blasted, spread itself, regularly, over the surface and quickly covered an area several hundred times the area of the initial spot. Oil-spread was also accelerated by slight temperature rise. Further, it was found that this oil film could not be removed by rubbing for any length of time with clean wipers. Removal was only possible by the use of a solvent or by raising the temperature of the metal high enough to burn off the oil.

Although at the present moment the functioning of this oil film is not quite perfectly understood, the fact remains that vapour blasting provides an oil retaining surface that is most effective for forming conditioned surfaces on cylinder walls. The treatment has been applied to steel liners after insertion and honing, and was accomplished by means of a specially designed machine which was capable of providing uniform blasting over the whole cylinder wall. (1947)

Tungsten Carbide Tools

A short series of lectures has been arranged by Croydon Education Committee at the Croydon Polytechnic, Mechanical Engineering Department, dealing with the manufacture, design and application of tungsten carbide cutting tools. The lectures will be given on Mondays, from 7 p.m. to 9 p.m. during the Session 1950-51. The dates are as follows:—

Monday, 9th April
Monday, 16th April
Monday, 23rd April
Monday, 30th April
Monday, 7th May
Monday, 21st May.

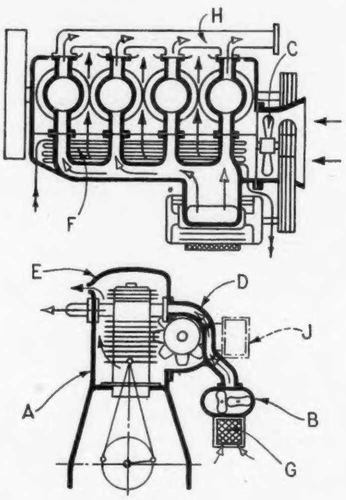
The lecturer will be F. H. Bates, A.M.I.Prod.E., Chief Education Officer, A. C. Wickman Ltd. (Wimet Division), Coventry. The fee for the course is one guinea, which should be sent, together with full name and address of the intending student, before the date of the first lecture. Communications should be addressed to the Secretary, Croydon Polytechnic, Scarbrook Road, Croydon.

CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Pressure-charged Air-cooled Engine

BOTH cooling air and charging air are by this arrangement delivered to a multi-cylinder engine by means of a one-piece cowling, flanged to the casing enshrouding the cylinders. In the example shown, four individual cylinders are secured to a crank case and surrounded by a jacket A, integral with the crank case. Combustion air is supplied by a Roots' blower B and cooling air by a rotary blower C, both driven from the engine crankshaft by V-belts. The ducting for the two blowers is embodied



No. 637003

in a cowl D which completes the cylinder enclosure. Cooling air from blower C flows into the enclosure, over the finned cylinders and out through three exit-apertures E.

Tube coils F for a lubricating oil cooler are housed in the lower run of the cowling. Drawn through an air filter G, the charging air from blower B is manifolded to the four cylinder intake ports and on the opposite side of the engine the exhaust ports are connected to exhaust manifold H.

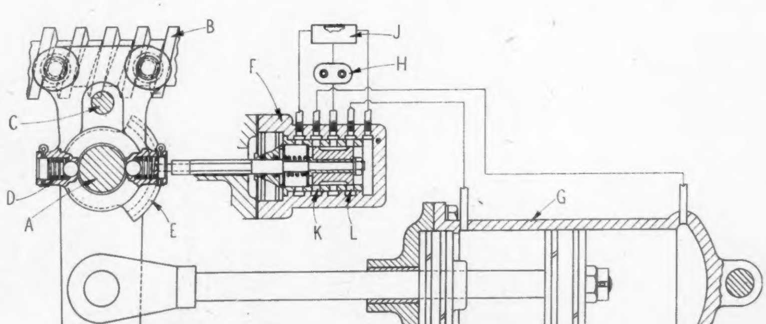
In a normally-aspirated engine an air filter may be fitted, as shown in dotted outline at J. *Patent No.* 637003. *Schweizerische Lokomotiv-und Maschinenfabrik* (*Switzerland*).

Power-Assisted Steering Gear

INSTEAD of being integral with or rigidly attached to the rocker shaft A, the actuating member carrying the two coned pins engaging cam B on the steering shaft pivots about a pin C on the rocker shaft arm. The lower end of the actuating member loosely encircles the rocker shaft and is normally held in a neutral position intermediate the limits of movement by diametrically opposite balls loaded by helical springs D. Also formed on the actuating member is a segment E with a concentric groove from which connection is established with the stem of the hydraulic valve unit F controlling the booster cylinder G pivotally mounted on the vehicle structure. The piston of the booster is linked to the drop arm in the usual manner. Completing the hydraulic equipment is a gear-type pump H and a fluid reservoir J.

With the actuating member in the neutral position, as shown, the pump delivers fluid to the mid-position of the control unit, whence it escapes by by-passing the end surfaces of the valve, by reason of the axially wider ports K and L, to the ends of the unit and is piped back to the reservoir. As soon as sufficient torque is applied to the actuating member by the steering cam to exceed the spring loading of one of the balls abutting the rocker shaft, the valve is moved to shut off admission to one end of the control unit and all the fluid under pressure is then delivered to the appropriate end of the booster cylinder.

It will be noted that should the effort tending to swing the drop arm originate from road wheel reaction instead of from rotation of the steering shaft, the booster will assist the driver if sufficient opposing torque is applied to overcome the pre-loading of the balls. *Patent No.* 635483. *Ross Gear and Tool Co.* (*U.S.A.*).



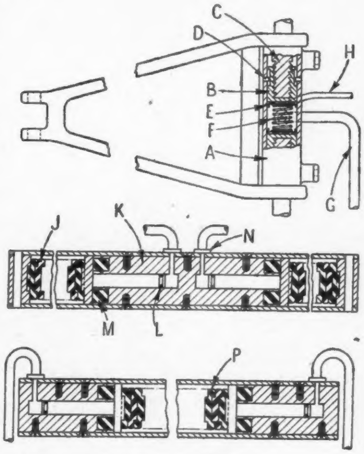
No. 635483

Suspension System

IN this system independently arranged wheel mountings are in fluid connection with resilient device of the rubber spring type remotely located on the chassis frame. At each wheel mounting a laterally disposed wishbone arm pivots about the axis of a cylinder A attached to the frame. In this cylinder operate two pistons B internally screwed with multi-start buttress threads, respectively of opposite hand. The pistons, mounted on axially aligned screwed spindles C secured to the arms of the wishbone, are prevented from rotation by keys D in the cylinder engaging longitudinal slots. Rubber cup washers E are provided on the face of each piston and supported by a common spring F. Pipe lines G and H establish respectively connection with a chassis-mounted spring unit and with the control equipment by which the static pressure may be adjusted.

Two types of spring unit are described, being intended respectively for independent or interdependent suspension of a pair of road wheels. For the former, a rubber spring J of the nested disc type is housed in each end of a tubular casing. Each is placed in compression by means of a plunger K slidable in a bore in a centrally disposed body. A rubber sealing ring L is provided near the end of the plunger and a resilient seating M for the head of the plunger is mounted in the end of the body. Connection with the pressure chamber is by pipe line G, past a damping device N permitting a free flow of fluid into the chamber but only a restricted flow outwards.

Similar component parts are used for the spring unit for an interdependent pair of wheels. In this case, however, the plungers are located at the ends of the casing and operate against a common



No. 634115

centrally disposed rubber spring P. It will be noted in this instance the rubber discs are arranged in opposition and at the centre an annular supporting ring is fitted.

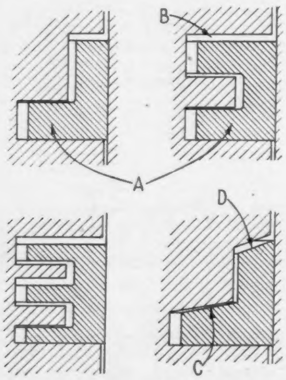
Front wheels are connected to one spring unit and rear wheels to a second unit. Each of these systems is separately connected to a control circuit which includes a fluid reservoir and a handpump so that the static pressure in either system may be adjusted to meet varied requirements. *Patent No.* 634115. *Dunlop Rubber Co., Ltd., and R. M. Seddon.*

Piston and Ring Assembly

THE phenomenon of "ring flutter", resulting in excessive "blow-by" is held to be not a simple vibrating movement but a radial collapse of the ring during a definite period of each operational cycle of the engine. Under the influence of inertia forces the clearance between the upper surfaces of ring and groove is taken up and the path for gas flow to the inner surface of the ring is interrupted.

To circumvent this problem, contiguous surfaces of ring and groove are disposed intermediate the upper and lower surfaces. These serve for the location of the ring in the groove and enable clearance to be maintained at the upper face whereby gas pressure may reach at least an adequate proportion of the inner surface of the ring to prevent a radial collapse.

Four types of ring are proposed, being respectively of L-, U-, W-, or wedge-shaped cross section, fitted in stepped piston grooves of corresponding section. In each the axial location of the ring is determined by the lower limb A of the ring while adequate clearance is provided at B above the ring. In the case of the W-section ring such clearance is arranged above both the upper and the central limbs. For the wedge-shaped ring it is claimed that most satisfactory results are obtained if, relative to a plane normal to the piston axis, the angle of lower surface C is 10 deg. and that of surface D 20 deg. *Patent No.* 636286. *P. de Kantzow Dykes and The Motor Industry Research Association.*

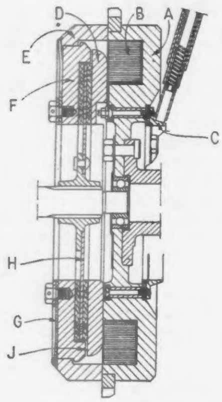


No. 636286

Electro-Magnetic Friction Clutch

THE object of this invention is to permit the use of non-metallic friction facings on the driven element whilst permitting the virtual closing of the magnetic flux outside the diameter of the driven disc. Flywheel A carries in an annular recess an earthed exciting coil B connected to an insulated slip ring C. To the face of the flywheel is secured a ring D and a peripheral flange E. Axially slidable in E is a presser ring F resiliently supported and keyed against rotation by a series of spring fingers G each engaged in a radial slot in the face of member E.

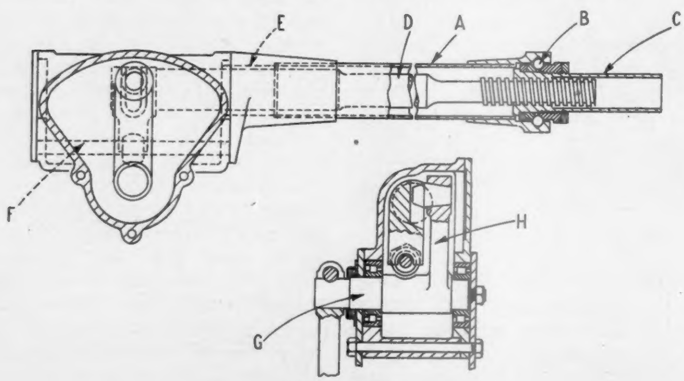
The presser ring F is recessed to receive a conventional, friction material faced clutch disc H slidably splined to the driven shaft. It will be noted that rings D and F closely approach one another when the clutch is energized, leaving only a small air gap J corresponding to the clearance necessary for the axial movement of the presser ring to produce a clamping effect for a specified extent of wear of the facings of the driven disc. *Patent No.* 636267. *R. S. de Lavaud* (*Brazil*).



No. 636267

Tractor Steering Gear

ALTHOUGH particularly suitable for agricultural tractors this gear may also be applied to other land or road vehicles. A hollow stationary column A is furnished with a thrust bearing B to support a manually rotatable hollow member C on which is mounted the steering wheel. Within the column is housed a shaft D, axially slidable in bearing E, the screwed outer end of which is engaged by member C. The inner end of the shaft is formed with a lateral extension which has a sliding connection, through a hollow ball, with a guide rod F mounted in a housing attached to the inner end of the column.



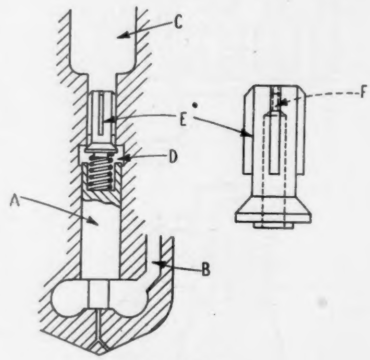
No. 637069

In the housing is mounted a cross-shaft G having an arm H carrying a laterally projecting frusto-conical pin which engages a groove formed in one side of the sliding shaft arm.

In a modified construction the sliding shaft is formed with a head extending laterally on both sides in order to impart opposite angular movements to a pair of coaxial cross-shafts mounted one on each side of the sliding shaft. *Patent No.* 637069. *Burman & Sons, Ltd., and W. H. Briggs.*

Hydraulically Damped Fuel Injector

IN fuel injection nozzles in which the needle is lifted by fuel pressure, regularity of operation may be disturbed under high speed conditions. It is thought that this is due principally to the sudden increase in volume of the space available to fuel under pressure, producing interference in the pressure-wave system between the fuel pump and the injector. This invention seeks to minimise such disturbance by preventing sudden opening of the needle, while retaining the desirable rapid closure.



No. 637038

Fuel is supplied to a space around the end of needle A by way of a drilling B in the body. The needle is urged downwardly to close the spray hole by the pressure of liquid in a reservoir C, which is in communication with a source of liquid pressure, not shown. A bore between chamber D at the head of the needle and the reservoir is controlled by a valve E. In this valve is an axial bore terminating at the upper end in a restricted orifice F. A helical spring, seated in a recess in the end of the needle, holds the valve E up to its seat.

So long as the needle is in the closed position chamber D is subjected to the same pressure as obtains in reservoir C. When the pressure in the fuel supply duct exceeds a specified value the needle is lifted and the pressure in chamber D increases rapidly and acts on the needle to oppose the opening movement. As the fluid in the chamber D can only escape to the reservoir by way of the restricted bore in the valve the needle is lifted relatively gradually, as compared with the usual type. As fuel supply pressure drops the pressure in the reservoir will lift the valve and effect a rapid closure of the needle.

Oil for the reservoir may be obtained from one or more cylinders of the fuel injection pump through a reducing valve, and the specific pressure may be regulated as a function of engine rotational speed. *Patent No.* 637038. *N.V. Bataafsche Petroleum Maatschappij* (*Holland*).